SEP 10 1941

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September, 1941

Heating · Piping and Air Conditioning

gn • Installation
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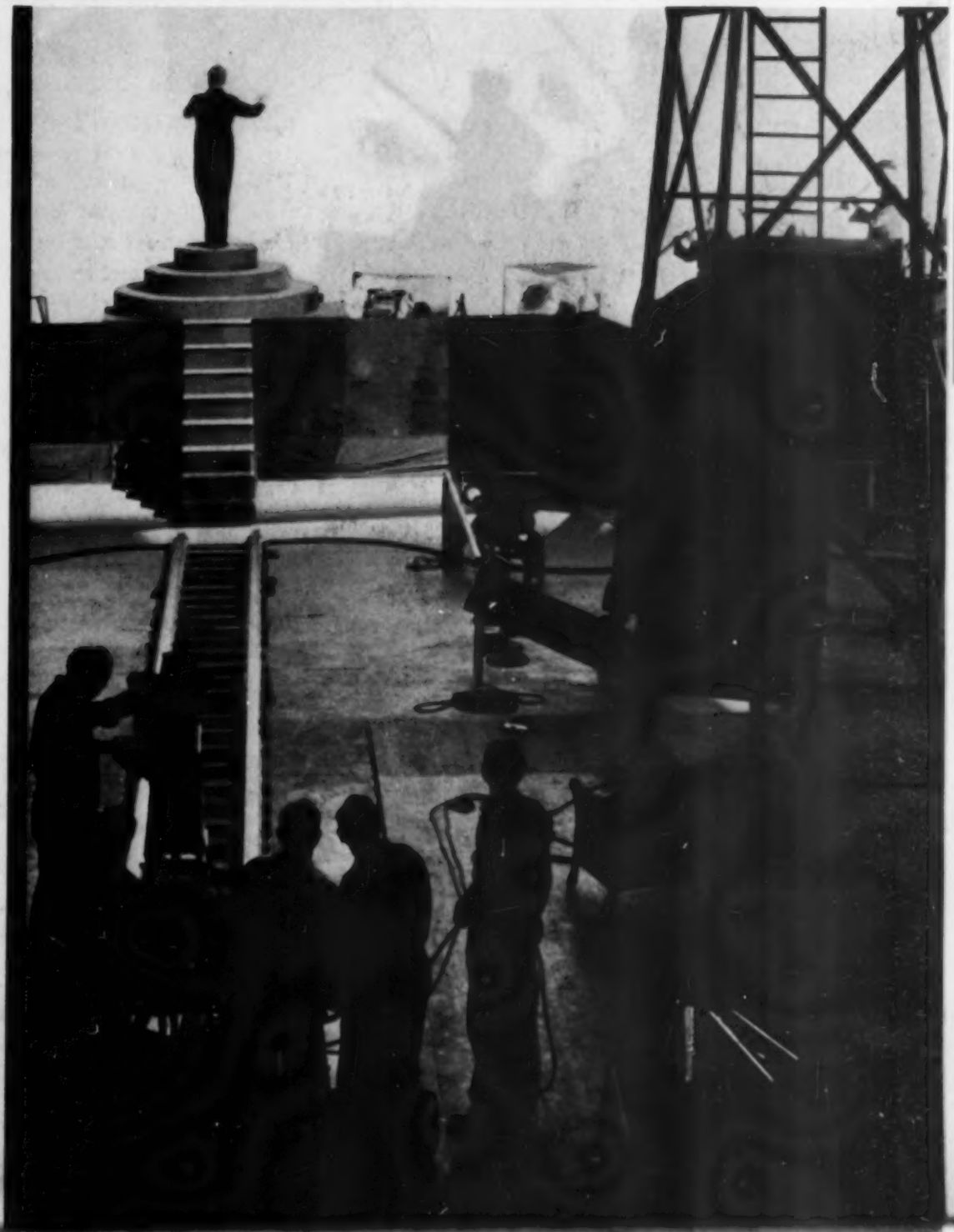
mercial, Industrial
Large Buildings

TROL of air temperatures, dities, cleanliness and factors is essential in the g of motion pictures—and er industrial processes too rous to mention. A brief ption on pp. 550-551 this emphasizes the impor- of air conditioning at the Disney studio, where the e shown at the right was . . . Other articles show ssential is air conditioning er industries—for exam- he manufacture of gages lefense item), as described 567-569. . . . In addition wealth of air conditioning resented this month, there any discussions of timely g and piping subjects



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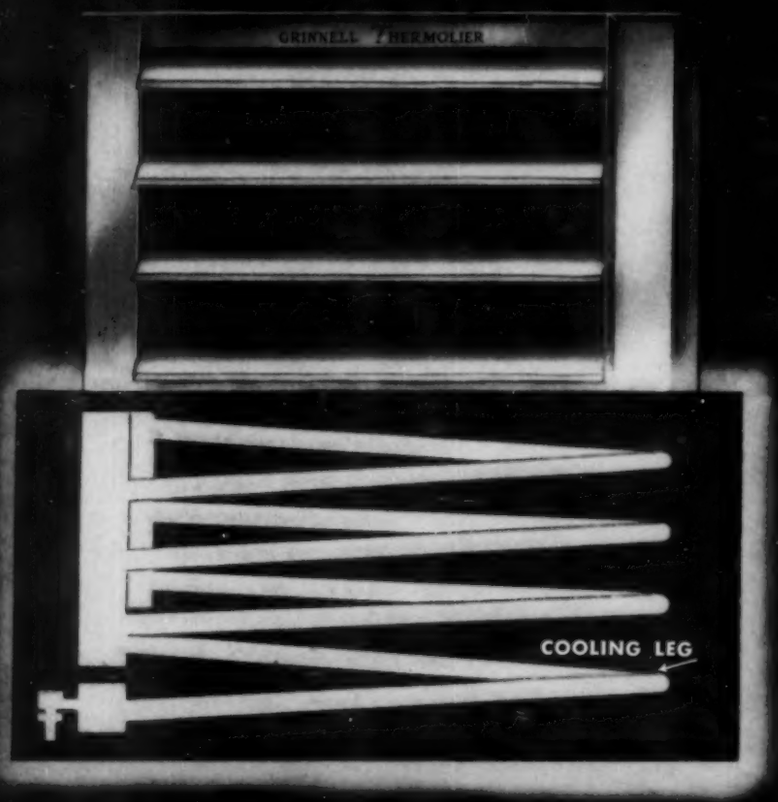
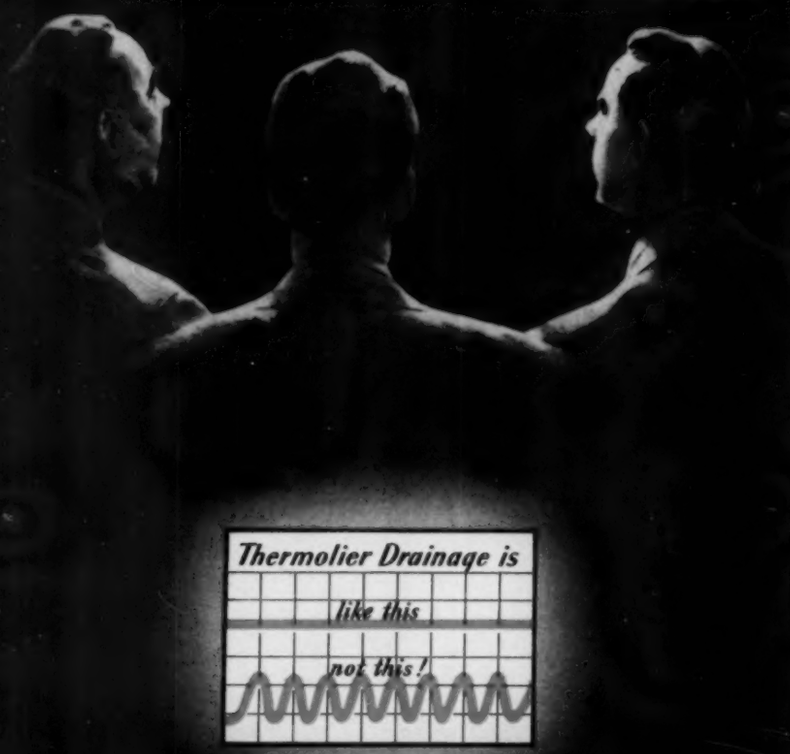
Doctors Don't Guess

Why Should You?





It takes more than a superficial examination to find the strength and weakness of Unit Heaters, too.

You're buying **HEAT** not ornaments, so don't stop at the housings (they're all smartly styled). *Look inside* . . . that's where the differences show up – vital differences in performance under actual *working* conditions, as opposed to standard ratings under ideal conditions.

Outstanding among all Thermolier features is the efficient *Internal* Cooling Leg, integral with the unit. It removes condensate *continuously* instead of intermittently; keeps *all* of the heater working all the time; eliminates "air binding" and hammer; permits use of simple thermostatic trap and is equal in actual cooling effect to *more than 100 feet* of exterior cooling piping.

This one feature *alone* places Thermoliers ahead of all other unit heaters in heating efficiency. Send for complete Data Book explaining Thermolier features. Grinnell Company, Inc., Executive Offices Providence, R. I. Branch offices in principal cities of the United States and Canada.

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2. *Positive Built-In Drainage* – every tube is pitched for complete drainage of condensate.
3. *Superior Fin Design* – square fins instead of round – 24% more radiating surface. Patented collars need no solder for strength. Dirt collection is reduced to a minimum.

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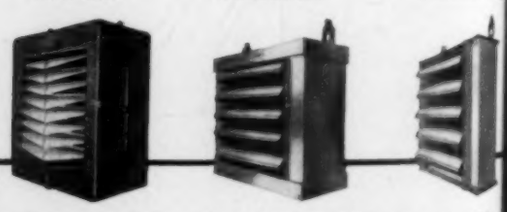

GRINNELL

THERMOLIER

THE UNIT HEATER WITH 14 POINTS OF SUPERIORITY

The Editor's Pages

OPACS ALLOCATES FREON REFRIGERANTS

Allocation of available supplies of "Freon" refrigerant gases to users and manufacturers of civilian refrigeration and air conditioning equipment in the order of relative importance to the public welfare is directed in a program announced last month by the office of price administration and civilian supply.

"Freon" is the proprietary name for several types of chlorinated hydrocarbon refrigerants produced only by Kinetic Chemicals, Inc., a du Pont subsidiary. These are used in several types of domestic refrigerators, as well as in commercial and industrial equipment and in all types of air conditioning equipment, including railroad cars. They are manufactured from carbon tetrachloride, which in turn is made from chlorine, and heavy defense needs for this basic chemical have caused a shortage in many of its derivatives.

A senior classification is assigned in the program to maintenance of all types of refrigerating equipment now operating, and existing air conditioning equipment in hospitals, clinics and sanatoria requiring these refrigerants.

Maintenance of industrial air conditioning equipment already installed ranks next in preference, followed by maintenance of other air conditioning equipment, and then by manufacture of new refrigeration and air conditioning equipment.

Current supplies are expected to be adequate for the maintenance of all installed refrigeration and air conditioning equipment, but some deliveries for new units may have to be deferred until the summer ice cream and air conditioning season is passed, it is believed. OPACS suggests that (1) reasonable requirements should be anticipated, but shipment of more than a 30 days' supply should not be requested; (2) empty cylinders should be returned promptly; and (3) pooling of orders by separate units of one establishment should be undertaken where feasible to reduce the number of cylinders required and to keep inventories from becoming excessive. The program, which will expire December 31, 1941, unless sooner terminated, will be administered by the division of priorities of the office of production management.

* * *

Stokers—Factory sales of mechanical coal stokers for the first six months of this year totaled 67,308 units of all sizes and types compared with 37,467 machines for the corresponding period in 1940, or an increase of 79½ per cent, the "committee of ten" points out. These figures are based on reports furnished to the department of commerce by 76 manufacturers of the 101 on the list. Shipments of oil burners and oil burning units of all sizes in the same period in 1941 increased 53½ per cent over the first six months of 1940.

* * *

WAR CONDITIONING BOOSTS INTEREST IN EXPOSITION

Leading manufacturers of heating, ventilating and air conditioning equipment are making active preparations to coöperate in the Seventh International Heating and Ventilating Exposition, in Philadelphia, which has taken on greatly increased importance because of the urgency of the national "war conditioning" effort, according to Charles F. Roth. The exposition will be held in the Commercial Museum from January 26 to 30.

It is a biennial event, held under the management of the International Exposition Co., of which Mr. Roth is president, and early bookings are the rule. This year the attitude of a majority of exhibitors is reflected by the fact that as of September 1, 80 per cent of the available space has been allocated.

Coincident with the dates of the exposition, the American Society of Heating and Ventilating Engineers will convene its 48th annual meeting. The National Warm Air Heating and Air Conditioning Association and other organizations are also planning to hold meetings at the same time.

CHANGES NAME TO INDUSTRIAL HYGIENE FOUNDATION

The trustees of Air Hygiene Foundation of America, Inc., have announced the change of the organization's name to Industrial Hygiene Foundation of America, Inc., in order to more clearly describe the foundation's expanding activities and membership services for the protection of employee health. When the foundation was organized emphasis was placed on silicosis, No. 1 industrial health problem at that time. With the rapid advances of industrial hygiene and the multiplication of new occupational health problems, the foundation's work has steadily broadened.

A study is now underway, with the collaboration of the U. S. Public Health Service, to help reduce sick absenteeism, which is said to be costing the heavy industries alone about one billion manhours yearly. Other accomplishments include continuing studies of x ray techniques for large scale physical examinations in industry; studies of the control of toxic fumes and gases, including proper precautions in welding; research to develop further practical data on exhaust ventilation for employee health protection; and investigations of "protector" dusts, including aluminum, to combat dust diseases.

GOVERNMENT WANTS INDUSTRIAL SPECIALISTS

The government continues its search for specialists in all branches of industry and business. The federal civil service examination for industrial specialists, announced on July 7, has been amended to remain open for receipt of applications until further notice.

The national defense program needs men with experience in one or more of the following fields: iron and steel; non-ferrous metals; machine tools; ordnance; aircraft, marine and automotive equipment; railroad repair shops; radio and

other electrical equipment, supplies and apparatus; textiles; forest products; paper; printing and publishing; chemicals and allied products; plastics; petroleum and coal products; rubber products; stone, clay and glass products; leather and its manufactures; and food and kindred products.

Salaries range from $2600 to $5600 in the various grades. No written examination is given. Application forms may be obtained from civil service representatives at any first or second class postoffice or at a civil service district office.

HENDERSON CALLS EXCISE TAX UNFAIR

In a statement to the finance committee of the United States Senate on August 19, William B. Henderson, executive vice-president of the Air Conditioning and Refrigerating Machinery Association, said that the revenue bill of 1941 (HR 5417), as passed by the House of Representatives, contained a substantial inequity and an unfair discrimination against the users and manufacturers of refrigerating and air conditioning machinery by proposing a 10 per cent excise tax on the sale of all refrigerating and air conditioning machinery.

Few items of mechanical equipment are as essential to the national welfare and to the national economy as the refrigerating machinery used for commercial and industrial purposes, said Mr. Henderson. Controlled temperatures and humidity are required for the processing of hundreds of products in common daily use. To illustrate: Controlled temperatures and humidity are essential to the output of many chemical products; of most drugs, medicines, serums and vaccines; of explosives; of petroleum products (particularly high octane aviation gasoline); of synthetic products such as nylon, rayon, and neoprene; of precision instruments, range finders, bomb sights, airplane instruments, optical goods and many other products.

Important though refrigerating and air conditioning machinery is to American industry, refrigeration is of indispensable importance in maintaining and protecting the health of the civilian population and the armed forces of the United States, he declared.

* * *

Indianapolis welcomes a nationally known firm this month—the Hoffman Specialty Co., formerly of Waterbury, Conn., is moving its general offices, research laboratories and manufacturing facilities to its new office and plant at 1001 York St. The move is announced to be in line with the redistribution of eastern manufacturing facilities to plants in the middle west, due to the emergency demands of the defense program. Paul G. Hoffman, chairman of the board, is also president of Studebaker Corp., South Bend, Ind. . . . . Warren Webster Jr., president of Warren Webster & Co., has announced the appointment of D. F. Bennett as manager of the radiation and unit heater division of the sales department. Mr. Bennett replaces Clarence E. Scott, who resigned in order to accept a position as sales manager of the air conditioning division of Fedders Mfg. Co., Inc. . . . . H. W. Sweatt, president of the Minneapolis-Honeywell Regulator Co., announced last month the addition to the executive staff of Alfred M. Wilson, as assistant to the president.

AIR CONDITIONED ORDNANCE PLANT

Construction of the Gadsden, Ala., ordnance plant, which will forge and machine shells, began November 1, 1940, and was 97 per cent complete, according to the war department, on August 25. The construction force of more than 400 men has erected 23 buildings, installed three miles of road, three miles of highway and fenced in the 390 acre reservation.

This plant is said to be the largest single air conditioned factory unit in the world. The normally high temperature in that region added to the heat generated by hundreds of machine tools and motor shops made air conditioning a virtual necessity. The 150,000 sq ft machine shop is lighted by three miles of flourescent light.

BLAST FURNACE CONDITIONING A DEFENSE MEASURE

Government officials and steel industry executives concerned with increasing steel production for our national defense effort have been studying the advantages of the wider application of air conditioning equipment to blast furnaces in recent weeks. A number of such installations for dehumidifying the tremendous quantities of air required have shown important savings and increased capacities, and it is felt the wider use of blast furnace air conditioning may be the answer to a vital defense question.

The results achieved with air conditioning in this field show up more favorably in the South, because of higher prevailing humidities, but worthwhile economies and capacity increases can also be achieved in northern plants.

STERILIZED AIR

The first large scale application of bactericidal ultraviolet to air conditioning in an industrial plant was completed recently at the Hillside, N. J., plant of Bristol-Myers Co.

All air passing through the air ducts at the rate of 40,000 cfm is rendered 95 per cent free of air borne bacteria by 104 sterilizing lamps. The 30 in. lamps are mounted vertically a foot apart, in rows of 15 in. apart, in a special chamber. Each lamp is operated at 475 volts and 0.04 ampere, obtained from a current regulating transformer having an open circuit voltage of 3000 volts and supplying four lamps in series. Twenty-six transformers are required for the 104 lamps (Photo courtesy Westinghouse.)

Heating · Piping and Air Conditioning

with which is merged the "Aerologist"

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Operation • Maintenance

Commercial, Industrial
and Large Buildings

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The Nash Vapor Turbine has but one moving part, rotating in the casing without metallic contact, and requiring no internal lubrication. Quiet, compact, and trouble-proof. Bulletin A-290 is free on request.

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TIFFANY & CO.'S NEW AIR CONDITIONED BUILDING ITS FEATURES DESCRIBED

UNUSUALLY INTERESTING DESCRIPTION OF THE MAGNIFICENT STRUCTURE NOW HOUSING THIS WORLD-FAMOUS CONCERN

TIFFANY & CO., jewelry, silverware and stationery house, moved last fall into its new air conditioned building at Fifth Ave. and 57th St., New York City. An air conditioning system for a building serving the many and varied uses of this one has much of interest. The system is divided into six independent zones. Chilled water from a steam turbine driven centrifugal refrigeration machine is pumped to the sensible and latent heat coils. The seventh floor is entirely devoted to manufacturing, and required special treatment. Careful consideration was given to air diffusion, control, prevention of downdrafts over large glass areas in cold weather, and to numerous other phases of the air conditioning and heating design. These are all explained

LAST October, Tiffany & Co., world-famous jewelry, silverware and stationery house, moved 20 blocks north into its new eight story building on New York City's Fifth Ave. at the corner of 57th St.

The air conditioning system (designed by Jaros, Baum & Bolles, consulting engineers) is sub-divided into six independent horizontal zones—separate systems with independent supply and return ducts—thus taking account of differences in exposure to sun heat and the loads involved. The first zone provides for the main floor and mezzanine (supply fan capacity 28,000 cfm), the second zone for the second and third floors (34,000 cfm), the third zone for the fourth and fifth floors (26,000 cfm), the fourth zone for the sixth floor (17,000 cfm), the fifth zone for the seventh floor (17,000 cfm), and the sixth zone for the basement (10,000 cfm).

By Lester E. Kelley, Tiffany & Co.

In all lower headroom areas, below hanging ceilings and in small conditioned spaces, high velocity, aspirating type ceiling air outlets are used, as illustrated

About 70 per cent of the main floor is occupied by the main sales area, comprising the diamond, gem, ring, jewelry, watch, gold, silver toiletware, stationery and watch repair departments. It is a room 80 by 100 ft with no obstructing columns. The ceiling, 23 ft, 10 in. high, is beautifully designed in a square pattern which artistically conceals some lighting units as well as the air distribution grilles and ducts, all hidden within and accessible from the attic space between this floor and the mezzanine.

There are five large show windows on the two street fronts, two toward the west on Fifth Ave. and three toward the north on 57th St., as well as two revolving door entrances, one on each front. Above the windows and doors large glass areas occupy a substantial part of the two side walls.

Air Leakage Problems

Due to the revolving doors and the elevator shafts at the east side of this large area there was a chance of the development of cross-currents, and it was necessary to deliver air into the area in such a manner as to offset this possibility. This necessity and the character of the architectural design of the very high ceiling led to the development of a particular design for the 45 ceiling grilles at the outlet openings of the attic duct work. In a frame 1 ft square and 3 in. high, two series of

To offset the possibility of cross-currents, and because of the architectural treatment of the high ceiling, a special design was worked out for the ceiling grilles in the main floor sales area, as shown here

fins are arranged in two horizontal levels and perpendicular to each other. The entire frame can be lifted by a handle and turned at 90 deg, and in combination with the possibility of changing the angle of discharge, this allows for a wide variety of air distribution.

To overcome the problem of leakage at the large show windows and lights of glass in winter, two recirculating booster heater systems, one for the west and the other for the north front, were installed. The air for both is taken from the conditioned area of the main floor. By means of duct work the delivered air is carried to the five individual sales windows and the seven large lights of glass. On the discharge side of these heating units booster coils are installed, thus creating a veil of conditioned air in front of the large glass panes.

The problem of leakage at the revolving doors was dealt with in a different manner. Two quadrants of the door when at rest are supplied with a discharge of warm air. At the same time, on each side of the revolving door, grilles discharge a curtain of warm air directly across the inner face of the door. When the door is in motion the leading quadrant discharges tempered air into the room, and the quadrant diagonally opposite, likewise tempered, gives the customer a feeling of warmth. Thus, neither the sales personnel nor the customer is affected by the objectionable drafts usually encountered near revolving doors.

Electrostatic Air Cleaners

Air for the main floor sales room is cleaned by electrostatic air cleaners, as is the air delivered to the mezzanine, first and second floors. The other air conditioning zones to date are equipped with dry type filters.

Tempering System for Mezzanine

As stated, the main and mezzanine floors are parts of the same air conditioning zone. The 9 ft, 6 in. high mezzanine consists of a number of small and medium sized rooms, particularly small private salesrooms, diamond offices, storage and file rooms, a pearl and diamond laboratory, small pearl stringing rooms, a darkroom, etc.

For conditioning the main floor, a low dew point is necessary. With the mezzanine, although for practical reasons belonging to the same zone, the requirement is different. To provide these smaller rooms with conditioned air adapted to their needs, a rather unique tempering system was evolved.

This tempering installation, together with the duct work for the main and mezzanine floors, is situated in the attic space above the main floor ceiling. Steam tempering coils might have been used for this purpose. However, this idea was abandoned in view of the delicate nature of the architectural treatment of the main floor ceiling, since steam pipe lines may be potential sources of trouble because of leaks, maintenance difficulties and heat radiation. Another means of control of the air supply to the mezzanine was therefore developed.

Sub-mastered, return duct, dry bulb thermostats in each of the three zoned systems of the mezzanine control three graduated motorized dampers in the air flow to and from the booster fans situated in the attic space. Rising room temperature automatically (1) reduces the amount of return air to the booster fan, (2) increases the amount of conditioned air to the booster fan, and (3) increases the amount of discharge air from the booster fan to the exhaust fan on the upper bulkhead—all of course in proper proportions. When the room temperature falls, the opposite action occurs.

Air Diffusion

The air diffusion in these mezzanine rooms is effected through high velocity aspirating type ceiling outlets, which are also installed in the second and third floor sales areas and the fourth and fifth cler-

ical and executive floors. Generally speaking, except for the special grilles used in the ceiling of the main floor, two ways of air delivery to the conditioned areas are used. Where long horizontal throws of air are required, side wall grilles are installed. In all lower headroom areas, below hanging ceilings, and in small air conditioned spaces, the aspirating type air diffusers are used. These diffusers draw in a volume of room air equal to about one-third of the volume of the primary supply air. Thus, the room air and supply air are mixed within the outlet before the whole air mass is discharged into the room, and this aspiration effect (which has nothing to do with air changes) causes the predetermined condition of the air to be established considerably above the breathing level. Therefore, greater temperature differentials are permissible, with accompanying economic advantages.

Other Zones

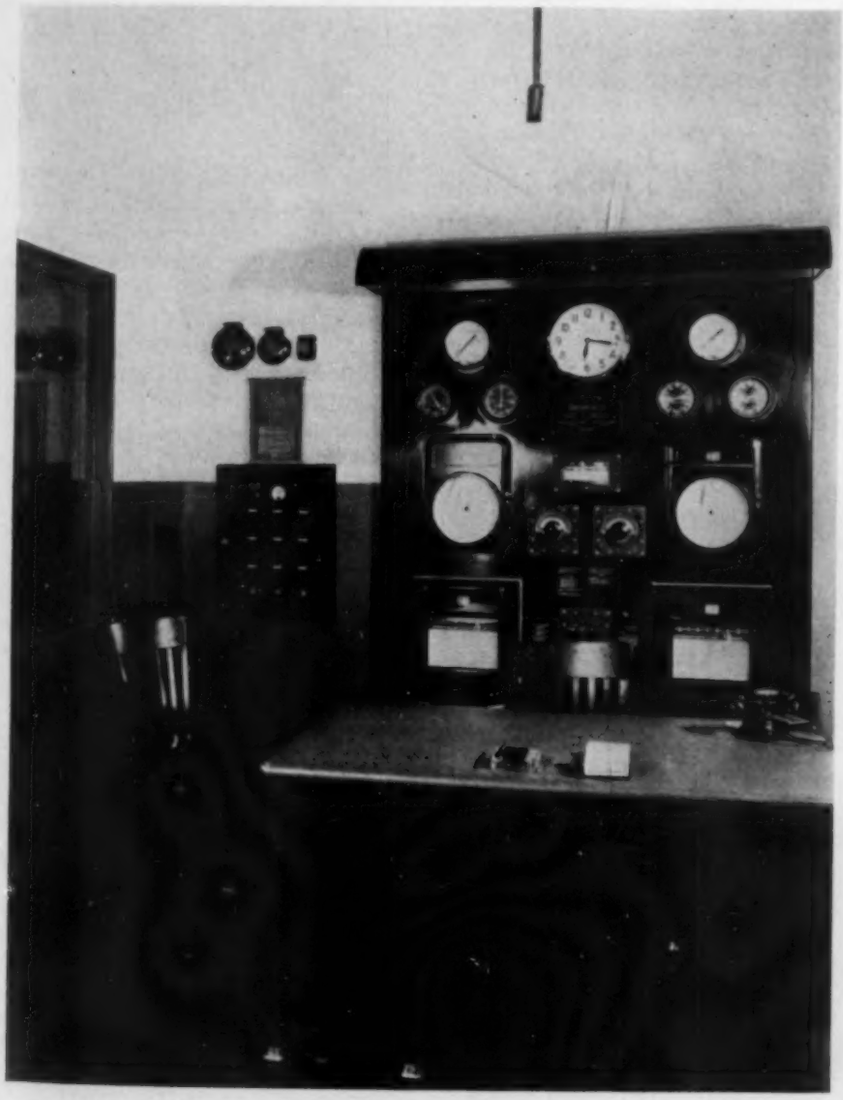
Architecturally, the sales rooms on the second and third floors are very much alike. As on the main floor, about 70 per cent of the floor area is given over to these sales rooms—silverware, leather goods, clocks and repair department on the second, china and glassware on the third floor. The ceilings are lower than on the main floor and the area is interrupted by six supporting columns.

These two floors are jointly conditioned by a separate zone of the system, and their treatment is alike, in contrast to the main and mezzanine floors. As mentioned before, air diffusion on these floors takes place through high velocity aspirating outlets, attached directly to the supply air ducts along the ceilings. To obtain a maximum amount of headroom, only the center portion of these areas, running east and west, was depressed to accommodate these ducts. The spaces inside these hanging ceilings serve as return air plenum chambers. They are accessible like the attic between main floor and mezzanine, and particular pains were taken to reduce air leakage to a minimum.

The next zone is that of the fourth and fifth floors. The lower floor embraces the auditing, record, tabulating and registry offices, silver and china reserve stockrooms, while the upper floor comprises the mail, bookkeeping, cashier and executive departments and the private offices of the executives.

For the control of the three zoned systems on the fourth floor—and that holds good also for the two lower floors just dealt with, and the sixth floor—there are sub-mastered room thermostats which control zone volume dampers in summer and zone volume dampers and modulating steam valves on the direct radiation during the winter. The fifth floor is more individually sub-divided—namely, into three larger zones and six private office rooms, the thermostats of the latter being manually adjustable.

Some of the controls and instruments



The Private Offices

In the private offices, situated on the street sides of the building and having very small occupancy, a particular condition had to be taken care of. The individual room thermostatic controls actuate volume dampers and modulating steam valves on the direct radiation. An unbalanced condition arises when excessive wind pressure and very low temperature outside strike the large windows. Quite often the room thermostats will be satisfied, which results in the steam valve closing and the volume control damper opening. When this occurs in one of these small offices the cold air rolling off the large windows may cause considerable discomfort to the occupant.

To offset this abnormal condition in spite of the room thermostat being satisfied, in each of these rooms an intermediate air stream thermostat has been installed, which is mounted directly on the under

side of the radiator grille, on the extreme outlet end of the radiator. As long as the room thermostat calls for heat the grille of the radiator enclosure is warm, its temperature being higher than the setting of the radiator thermostat. As soon as the room thermostat becomes satisfied, the steam goes off and the radiator grille cools down. The temperature of the grille drops rapidly because of the cold air rolling off the windows. When this occurs, due to the temperature setting of the radiator thermostat, the radiator steam valve is reopened and remains open until the radiator grille temperature becomes satisfactory. By this intermittent operation of the auxiliary thermostat the room conditions proper are in no way affected. That is, the difficulty is overcome by wiping the large glass exposure with a curtain of warm air.

Except for these private offices no thermometers are attached to the thermostats. The reason for entirely abolishing temperature indications and readings is that the scale of these thermostat thermometers is such that it is difficult for a layman to read, particularly on account of errors arising from different heights of the individuals. Moreover, quite often these thermometers are inaccurate. Thus, complaints might result based only on mistaken thermometer readings and imaginary conditions which could not be verified by psychrometer readings.

On the sixth floor, which is taken care of by a separate zone of the air conditioning plant, there are the clock and watch repair shops, some small offices, a photo room, and the designing and engraving department. This floor, moreover, is used for the distribution of all steam, water, air, gas, oxygen, etc., throughout the building, so that additional headroom had to be provided for pipe lines and ducts.

The Manufacturing Floor

The seventh floor, also conditioned through a special zone of the system, is the manufacturing floor; here are the jewelry shop, jewelry repair shop, the jewelry and silver polishing shops, as well as the jewelry designing department and jewelry shop office. Entirely given over to industrial purposes, this floor is provided with 100 per cent outside air, without any recirculation. The windows on this floor cannot be opened. Heating is done entirely by the air conditioning system to avoid intricate surfaces on which metal dust might collect. For conditioning purposes, this floor is sub-divided into six zones and one room (the plating room), each having sub-mastered room thermostats.

Two rooms on this floor—the silver and jewelry polishing shops—are equipped with polishing lathes which by law require a definite static pressure to be maintained at the extreme end of the duct work prior to entering the duct filters. The exhaust requirements have the tendency to unbalance the static pressure of the room proper. To overcome this condition it was necessary to provide separate means of exhaust and filtering from the lathe tables. The discharge from the lathe table exhaust, after being filtered, is returned to the polishing rooms in addition to the normal supply of conditioned air, with the result that the static pressure in these rooms remains nearly constant.

There are a number of electric hot plate heating stoves in the work shops on this floor which are used for heating alcohol and other chemicals. The resulting fumes are removed by separate exhaust fans above the stoves and an exhaust system not connected with the seventh floor main exhaust. To make elimination of these fumes foolproof, the electric circuit of each stove is tied in series with that of its exhaust fan, thus securing simultaneous starting of the fan and stove.

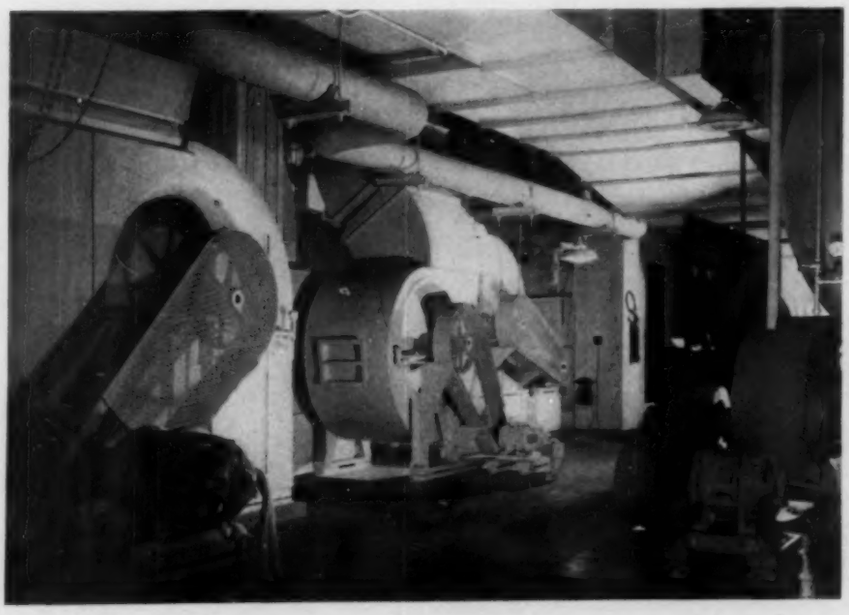
Toilet Exhaust

In addition to the horizontal zones of the air conditioning system there is one part of this system which takes care of a vertical section of the building—the toilet exhaust. The bulk of the toilets are in the southeast corner of the building. All toilet rooms are under 100 per cent exhaust into a common toilet exhaust duct. Outside air make-up is provided through louvered openings in the entrance doors, taking conditioned air directly from the respective floors. This arrangement eliminates the need of supply air ductwork and minimizes the amount of exhaust ductwork necessary.

The Mechanical Equipment

The mechanical equipment of this extensive and flexible air conditioning plant is situated partly in the basement and partly in the two bulkheads of the penthouse and on the roof.

The entire system uses chilled water as a cooling medium, supplied by a steam turbine driven centrifugal refrigerating compressor which has a capacity of 400 tons. The cooling effect for condensation is obtained from the cooling tower on the roof of the penthouse, the cool-



All the supply fans are arranged for variable speed, thus permitting change of air volume, depending upon the weather, and also flexibility of operation

ing water taking care first of the condenser of the refrigerating machine, then of the steam condenser of the turbine and eventually being returned to the tower.

The entire air conditioning system is under full automatic temperature regulation. In winter the incoming air is heated and humidified (with spray nozzles); in summer it is cooled and dehumidified by the cooling systems. The plant is so designed that 100 per cent outside air can be used during intermediate seasons.

Cooling and dehumidifying of the conditioned air during the summer is effected by chilled water being circulated through coils. The method of treatment of outside air and return air is rather unique. Two sets of coils are provided for each of the four zones from the main floor through the sixth floor; latent heat coils handle only outdoor air and sensible heat coils handle a mixture of outside and return air. These coils are arranged side by side, divided by steel baffles, having their water flowing in series, and each set of coils has a three way mixing valve controlling the amount of water.

Unusual Control Hook-Up

The summer control of these zone systems is interesting because of the unusual hook-up for control of latent and sensible heat. In addition to the common outdoor thermostat for all zones (and a separate one for the basement system), there are provided for each system one return duct, sub-mastered, dry bulb thermostat controlling the three way mixing valve in the chilled water connection to the sensible cooling coil, and one return duct, sub-mastered, wet bulb thermostat controlling the three way mixing valve in the chilled water connection to the latent cooling coil. Both thermostats are adjustable for temperature settings and for the inside-outside temperature variation ratio. This arrangement permits a wide range of chilled water temperature operation with corresponding decrease in the amount of water circulated to obtain this effect, which in turn reflects itself in a very effective operation of the centrifugal compressor.

All multiblade supply fans are arranged for variable speed, thus permitting change of air volume, depending upon the weather, and securing flexibility of operation. All air supply is discharged down from the roof to the respective zones through individual ducts, except the basement, for which a separate air intake some feet above street level is provided.

The conditioned air supplied from the penthouse bulkheads to the conditioned areas from the main floor through the sixth floor is carried down in a shaft common to all these zones and situated near the southeast corner of the building. This design confines the largest ductwork to the vertical riser and results in a reduction of the size of the horizontal ducts and therefore gives maximum headroom in the conditioned areas.

Part of the mechanical equipment is in the west section of the basement, the rest of that floor being occupied by shipping space. In the compressor room are the turbine compressor unit and the chilled water circulating pumps.

The Steam Room

All steam requirements are supplied by the New York Steam Corp.

The steam room contains all steam metering equipment and steam distribution headers, the hot water and preheating storage tanks, the basement air conditioning units, and the basement outside air and first floor booster units. While the ventilation supply for the compressor room is met by 100 per cent conditioned air, 50 per cent of the total air supply for the steam room is outside air, and the balance consists of conditioned return air from the basement—an arrangement which maintains excellent steam room temperature and humidity conditions.

Both these machine rooms are situated directly under the west part of the main sales area facing Fifth Ave.—certainly the most valuable sales space in the Tiffany establishment and probably in the world. It may be pointed out that to date not the slightest vibration emanating from any of the mechanical equipment or any noises from steam pressure reducing valve operation have been felt or heard on the main floor or anywhere else in the building.

The balance of the mechanical equipment is on the two bulkheads on top of the building. The upper bulkhead is occupied by the conditioned air supply and return units for the two zones comprising the main, mezzanine, second and third floors and the electrostatic air cleaners, through which the air passes before being discharged into these four floors.

The lower bulkhead, i.e. the lower floor of the penthouse, houses the conditioned air supply and return units for the next two zones which are comprised of the fourth, fifth and sixth stories, as well as the supply and exhaust units for the seventh floor, for which no recirculation is provided as pointed out before.

Cooling Tower Air

Considering the total volume of air handled when the conditioning plant runs at full capacity, the thermostatic control system is so arranged that a definite quantity of return air can continuously be spilled into the bulkheads, thus subjecting these areas to high static pressure. To relieve this condition an interesting method has been introduced. In the floor between the two bulkheads a large open type grating has been inserted, making these two levels common insofar as air pressures are concerned. In the roof of the upper bulkhead are two hand dampered outlets. One of these leads directly to the atmosphere and is used during the off cooling season. The other leads to the base of the 28 ft. copper lined, induced draft cooling tower. Operation of the 30 hp fan at the roof of the cooling tower results in the bulkheads being placed under static suction and the cooling tower receives the full benefit of the spilled air, which is at a wet bulb temperature lower than that of the atmosphere.

Architects for the building were Cross & Cross.

[Photos by Joseph W. Molitor, courtesy Anemostat Corp. of America.]

News about new and improved equipment appears in the department "Equipment Developments" published in HPAC every month.

BIRMINGHAM "SWATS THE SMOKE"

George H. Watson Describes Results of Birmingham's Smoke Abatement Program

THE first year of a city wide "swat the smoke" smoke abatement campaign in Birmingham, Ala., has accomplished definite results, among which has been the installation of improved firing equipment in a number of plants. In addition to the installation of underfeed stokers, some of the plants found other means also of eliminating smoke. The first to receive wide publicity was that in the Birmingham Electric Co. downtown steam plant, where blowers were installed for overfire air circulation to prevent smoke.

In a number of plants, notably laundries and dry cleaners, overfire steam jets were installed to draw in air over the firebed. Soft coal is the fuel used. The county courthouse, which uses coal pulverizers, found also that the installation of air vents over the firebeds eliminated smoke. This was done with the advice of the engineers who designed the heating plant.

Chemically Treated Coal

In still other plants the answer to the smoke elimination problem was found in the use of a chemically treated coal. This was the case at the $5,000,000 Elyton housing project which operates its own steam heating plant. Here a test was run with Commissioner of Public Improvements James W. Morgan, who has the smoke abatement program in charge, present. A report on this test as made to Commissioner Morgan by Daniel W. Bateman, technical adviser for the Birmingham housing authority, was as follows:

In order that you may have full information with regard to this test, we hereby submit the following information:

4210 lb of untreated coal was consumed over a period of 24 hr, which evaporated 36,437 lb of water at 183 F with an ash and clinker residue of 263 lb.

4210 lb of chemically treated coal was consumed over a period of 24 hr, which evaporated 40,250 lb of water at 183 F with a residue of ash and clinkers of 156 lb.

An analysis of these two results will show that 10.4 per cent more water was evaporated by the use of the treated coal.

This authority, as a result of this test, has instructed the contractor furnishing coal to treat all our coal in the future as the additional cost for the chemically treated is considerably less than the saving in fuel consumption.

We know that the use of this chemically treated coal will practically eliminate the smoke nuisance which you would normally get from a heating plant of this size.

Perhaps the most noteworthy installation to eliminate smoke was that in the Birmingham Electric Co. steam plant. This plant was erected in 1903-04, and served the city with light and power until 1914, when the load was taken over by hydroelectric plants. Since that time it has been used as a reserve plant and to supply the city demand for central station steam.

SWAT THE SMOKE is the battle cry in many an American city; results of the first year of the city wide smoke campaign in Birmingham, Ala., are highlighted here. . . . In addition to installation of underfeed stokers, a number of plants—particularly laundries and dry cleaning establishments—installed overfire steam jets, and chemically treated coal was found to give good results at a housing project (though some engineers contend that "a poor grade of coal remains a poor grade no matter what you do to it"). . . . An installation which received wide publicity was that in a downtown steam plant, where special blowers for supplying overfire air were put in

Overfire Blower Equipment

Existing equipment in this plant included six 600 hp boilers with chain grate furnaces. The problem was to provide some method of mixing air and gas above the fuel bed in each furnace, so as to cause complete combustion, thus eliminating high stack losses and smoke.

Company engineers and consultants decided upon the installation of two special 28 in. double width, 7500 cfm blowers, each serving three boilers. The blowers are mounted on the boiler superstructure and each is powered by a 30 hp induction motor.

The blowers discharge into plenum boxes which have a static pressure of 10 to 12 in. and thence through jets into the furnaces directly above the fuel bed. Each furnace has 20 jets of 1½ in. pipe. This overfire air is of greater static pressure than that through the fuel bed and hence cuts through the hot gas and air, causing thorough mixing and prompt combustion. No preheating of the overfire air was considered necessary.

It was necessary to redesign the arches of the furnaces somewhat before installing the jets. This redesign of the arches might in itself give the desired turbulence if it were possible to operate the plant at only one rating. However, most boilers have to operate over a considerable range of capacity, hence the need for supplementing the undergrate supply with the overfire jets. With these jets it is possible not only to

One of the six boilers in the Birmingham Electric steam plant, showing the 20 jets for overfire air

control the volume of air above the fuel bed, but its direction also.

This installation was praised by Commissioner Morgan as "the first big example of just what can be done to abate smoke in the city." It was made under the direction of C. S. Thorn, vice-president, and H. M. Killingsworth, superintendent of power plant and substations, of the Birmingham Electric Co.

In addition to these installations in some of the larger plants, an incessant campaign to encourage improved hand firing methods among home owners has been carried on. Commissioner Morgan in a statement estimated that 50 per cent of the early morning "smog" which often envelops the city could be eliminated by reversing the building of fires—that is, placing the coal on the bottom of the firebox with the wood on top.

The smoke abatement inspector, who incidentally has the power to arrest flagrant violators, conducted several schools for firemen and others, visited 5530 homes, 62 apartment houses, 104 plants, seven schools and eight hotels during the course of the first year of the campaign.

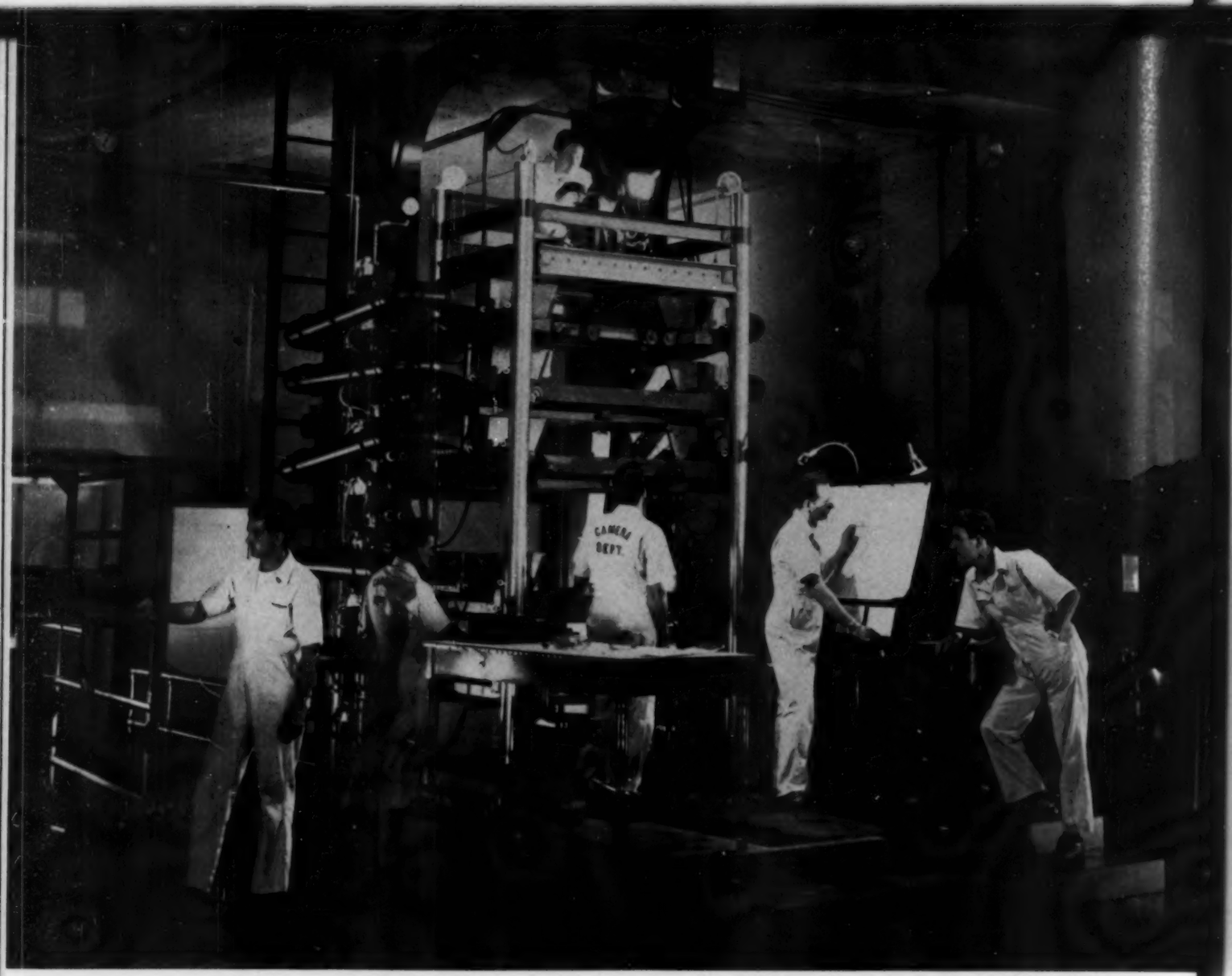
Commissioner Morgan emphasized that the city was not campaigning against soft coal, as Birmingham is in the center of an area which produces it, but he added, "We know from actual demonstration that soft coal can be burned with very little smoke."

FIGURING THE BUILDING VOLUME FOR FUEL CONSUMPTION STUDIES

A question that frequently comes up in connection with recording steam or fuel consumption for heating on a cubic foot of building volume basis is that of the proper building volume figure to use.

It happens that in most office buildings there is a figure of volume for the building already in use for some purpose or other. The figure may be the "architect's volume", which ordinarily includes the whole building from 6 in. below the surface of the basement floor up to and including all the roof structures—gables, penthouses, etc. For most buildings, there is also already in use a volume figure for tax purposes, and seldom do the two agree. If either such figure is already available for any given building, I think it is appropriate to go ahead and use it for the heating computations. If not, and if it is thereby necessary to compute a figure of volume for the building or buildings, I favor including all space which is heated from the heating plant either directly or indirectly. Thereby, the basement is included for it is heated to some extent or other, and ordinarily to a level practically the same as the building itself; and also the attic and other roof structures, since they are ordinarily heated greatly above outdoor temperatures by convection and by the stack effect of the building—this being true whether they have radiation or not. As a corollary, the outside surfaces of those parts of the building are giving up heat to the outside and this heat loss must continuously be made up. I believe that in the average office building the difference between the three possible figures of volume mentioned above would seldom be as much as 20 per cent, and would usually be less than 10 per cent.

No matter what the method used in computing the volume, it still enables comparison within the building of "before and after" heating economy and therefore is rational. The difference between the three possible volume determinations may exceed the values surmised above in the case of buildings other than office structures — notably, for instance, if the buildings were scattered, separated low buildings such as ordinarily compose a university or college group.—R. E. HIERONYMUS, general superintendent, Aldis & Co.



PIX BIZ LIX KIX—SAYS NIX TO OLD MAN WEATHER

. . . . That's the way "Variety"—trade paper of the amusement field noted for its contributions to the American language—might say that the motion picture industry avoids troubles by complete control of air conditions. Here are some facts about the Walt Disney studio which indicate the importance of control

AN IDEA of the importance of air conditioning to the motion picture industry is given by some of the statistics on the Walt Disney studio, near Burbank, Calif. Total conditioned floor space at present is about 250,000 sq ft. To do the job 40 million cfh of air is handled by 135 multiblade centrifugal fans, driven by 275 hp of electric motors. To make the air ducts required 600,000 lb of galvanized sheet iron and 40 sheet metal workers laboring 280 days.

To warm or cool the air, water is circulated by 18 electric pumps totaling 121 hp in addition to 300 hp of well pumps. There is three miles of water, steam, and "Freon" piping in the combined system, connected to more than 300 heat transfer coils which required 63,700 lb of copper in their manufacture.

Air temperature and humidity are regulated by 360 separate thermostats and humidistats, transmitting impulses by means of compressed air through 77,250 ft of copper tubing. To provide the necessary air cooling effect, the well water is supplemented by a 600 hp multiple compressor refrigeration plant. In all, there is more than 1300 motor horsepower needed for air conditioning functions.

The use of 10 to 15 outside air changes per hour in the buildings, with no recirculation, has been found highly satisfactory, as there is

***Above*—Photographers at the Walt Disney studio working with a special multiplane camera in a room where cleanliness and close control of temperature and humidity are essential to good results**

almost no trace of odor and the air is always fresh. In all buildings, cleanliness is highly important. In photographing a painting, dust would produce light effects that detract from the finished picture. Thorough filtering is also necessitated by the occurrence of severe dust storms in the valley, and by the presence of smudge pots in nearby citrus groves. The use of 100 per cent outside air obviously adds to the filtering job. The filters are of a viscous coated, crimped metal screen type, 2 in. thick. In the camera and cutting buildings a second set, using finer mesh, is provided for maximum dust removal. All of the 500 primary filters are interchangeable, and spare sets are kept on hand. Periodically they are cleaned under a jet of hot water, drained, dipped in odorless white mineral oil and again drained in racks that recover surplus oil. Disney engineers estimate the cost of cleaning filters at 2½c each, carting and replacing at 2½c, and the cost of oil and hot water 5c—or 10c each every two months. Air cleaning costs have averaged a dollar per year per thousand cubic feet of outside air supplied, and even this figure is being bettered as vacant spaces are planted with grass and shrubs.

The studio has an abundant supply of water in two wells, sunk 300 ft, with a combined capacity of 5200 gpm. The water temperature averages 67 deg and it can be used for precooling or heating air. About 3000 gpm is available for pre-conditioning. Banks of cleanable finned copper coils are situated directly after the filters in each fan room, six rows deep in the direction of air flow. Valve controls are so arranged that a fairly uniform temperature of about 55 deg is obtained for air leaving the coil when the outside air is below 55, and 75 deg whenever outside air is above this figure, thus reducing the maximum demands on the heating and refrigerating equipment. About 1000 gpm of water is available for the refrigeration condenser cooling, and the balance can be used for lawn sprinklers and service water. Two 150 hp submersible motor pumps supply water to mains and to a 150,000 gal tank, 100,000 gal being reserved at all times for fire protection. The remaining 50,000 gal space acts as a reservoir for the air conditioning water, which is maintained by automatic pumps equalizing the use of the two wells.

The studio has been fortunate in being able to dispose of this vast amount of water in a convenient manner. An old irrigation duct runs from the site of the studio to a basin near the bed of the Los Angeles River, and this is used. The city is contemplating the use of the water to beautify a section of an adjoining park, but meanwhile the neighborhood kids have discovered an excellent swimming hole, with a protective growth of shrubbery that has sprung up around the new oasis.

The animation building, center of the studio development and scene of much of Disney's current feature, The Reluctant Dragon, is constructed as a series of east-west wings connected to a central north-south corridor. It incorporates about 350 rooms, including artists' studios, picture review rooms, conference rooms, offices and service department work rooms, as well as a coffee shop and penthouse athletic club housing gymnasium, steam baths and showers, sun deck, rubbing rooms, and a lounge. There are 163 separate temperature control zones in this building. Pneumatic thermostats are generally set to hold 74 deg inside when it is 85 or below outside, and will automatically hold a higher room temperature as outside temperature increases, 78 deg being the maximum when the outside temperature is 100 or above. Temperatures can be maintained at thermostatic setting within plus or minus ½ deg. Individual humidity control is not provided for each zone, but is regulated for each wing by humidistats in the exhaust air duct.

[Photos courtesy General Electric Co.]

Standard crane camera equipment, showing air conditioning of the points where maximum temperature develops as result of lighting

INTERCHANGE OF POWER AND STEAM

P. W. Ross Reviews Results of Power and Steam Interchange Contracts Between Industrial Plants and Public Utility

THE beginning of contracts for interchange of steam and electricity between the Indianapolis Power & Light Co. and large industrial concerns in Indianapolis had its inception in an arrangement negotiated by the utility with Eli Lilly & Co. in 1931. At that time, the Lilly company, internationally known manufacturer of pharmaceuticals, was faced with the problem of renewing its own boilers or buying steam from the local electric utility. Steam was required by the Lilly company for the operation of steam turbines, air compressors, auxiliary equipment, heating and various process work. When the contract was made it was entirely conceivable and so recognized in the contract that the amount of steam required by the manufacturer, passing through the turbo-generator equipment in its plant, would produce more electrical energy before being drawn off as process steam than the manufacturing company could use.

It was, therefore, agreed that the customer's electrical system would be operated at all times in synchronism with the electrical system of the power company and at such times as the power generated by the customer exceeded his requirements, the power company would absorb in its system the excess energy thus produced. During those periods when the customer's electrical energy requirements exceeded that generated in its own turbines it would draw from the power company's system the deficiency amount. Steam at 175 psi pressure and 100 F superheat was to be provided by the utility at an anticipated rate of 60,000 lb per hr. All the steam delivered is metered at the customer's premises, and the customer is billed for all steam delivered at the utility's regular rate for industrial steam.

Since 1932, steam delivered annually to this customer has risen from 112,000,000 lb to 295,000,000 lb in 1940. The revenue to the utility for steam alone has risen from $41,000 in 1932 to $101,000 in 1940. In addition to this the kilowatthours purchased by the customer from the utility have risen from 1,788,000 in 1932 to 5,645,000 in 1940. Incidentally, it is interesting to note that there has been no feed back of power from the customer since 1936, and the utility's electric revenue has risen from $24,138 in 1932 to $62,745 in 1940. During the first five years of the contract the Lilly company returned to the utility electricity in the amount of $2673. In the first nine years of the contract the total revenue to the utility from this interchange contract has been a little over $1,000,000

High Pressure Steam Line

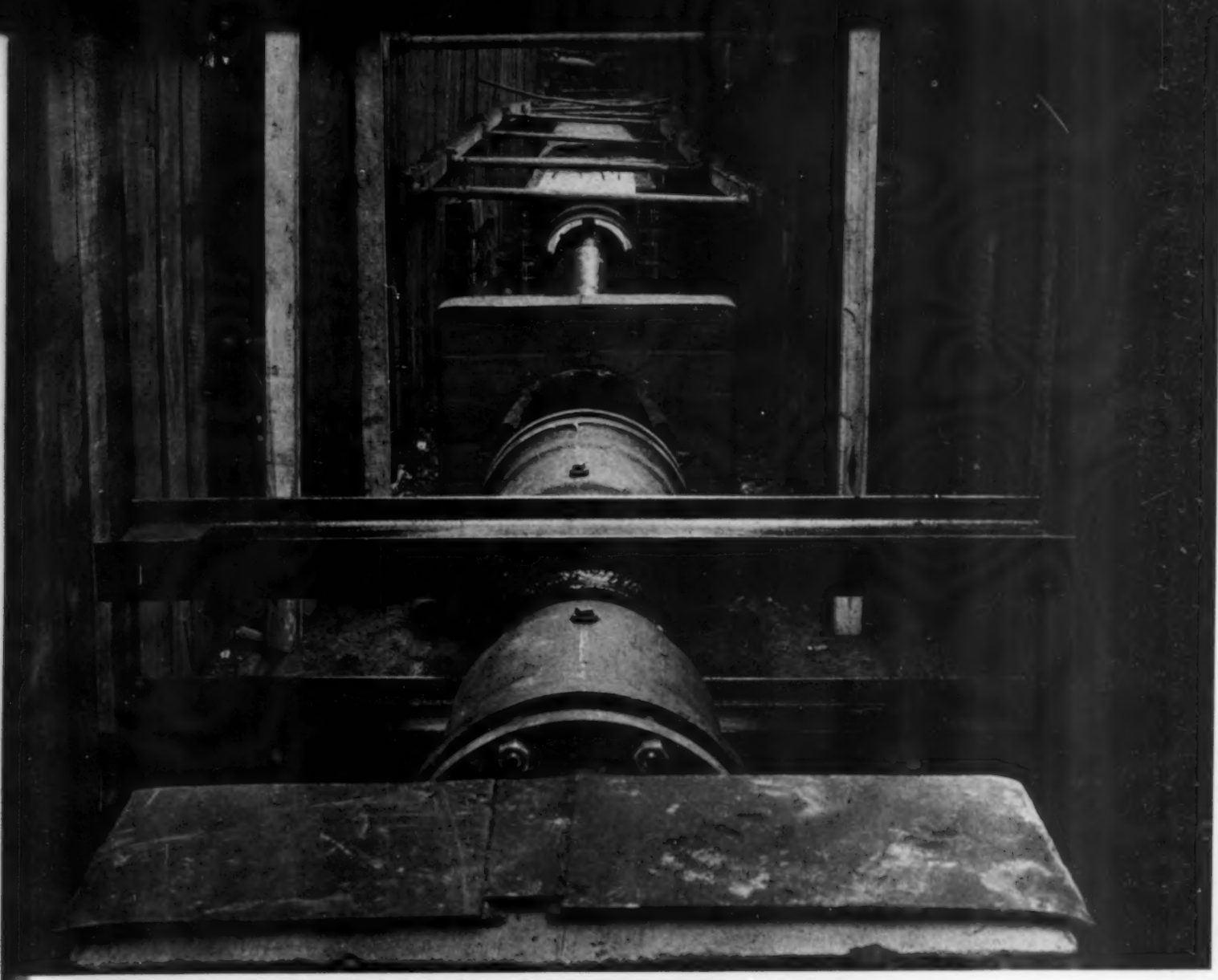
The Eli Lilly & Co. is situated approximately one mile from the Perry K generating station in the heart of the industrial district of Indianapolis. Sound engineering judgment dictated that a high pressure steam line of such length should have a considerable amount of reserve capacity in anticipation of other industries making similar use of high pressure steam. Therefore, the first 2600 ft was constructed of 16 in. pipe, the second 2600 ft being 10 in. pipe. The ultimate capacity of this line is calculated at 200,000 lb per hr of steam through the 16 in. part of the line and 90,000 lb per hr through the 10 in. section. With the large capacity of this 16 in. line it was deemed advisable to place a safety shut-off valve in the line where it leaves the boiler room. This valve will trip and close when the differential pressure through its orifice plate equals 5 psi. A sudden increase in the customer's steam demand does not affect this valve, because the volume of the line acts as a surge tank and smoothes out the sudden fluctuations of the load.

The wisdom of such an initial installation has been verified by the increased number of customers connected to it since its installation.

The first such customer was connected shortly after the line was installed and in operation when a contract was negotiated with the E. C. Atkins Co., nationally known saw manufacturer, to supply all of its steam and electric requirements. The conditions leading to the successful negotiating of this contract were similar to those of the Lilly company. In the Atkins plant the customer is operating an 800 hp corliss engine directly connected to an alternator which carries a portion of the electric load. This engine is operated only during the heating season, and the electric load is varied in accordance with the amount of steam required by the heating system. This customer purchases approximately 80,000,000 lb of steam and 2,250,000 kwhr annually.

During the year 1934, Kingan and Co., large meat packer, was confronted with an obsolete and inefficient boiler plant which was unable to furnish an adequate supply of steam. Under the method of operation, the overall power costs were excessive. An investigation of the economics of erecting a modern power plant, operating at approximately 400 psi, was begun. If the proposed plant had been erected the Indianapolis Power & Light Co.

INTERCHANGE — Back in May, 1932, William A. Hanley, director of engineering of Eli Lilly & Co. and member of HPAC's board of consulting and contributing editors, described in these pages the unique arrangement between his company and the Indianapolis Power & Light Co. for the interchange of steam and electricity. It is interesting to follow up Mr. Hanley's pioneering discussion of this subject with some of the information presented by Mr. Ross, assistant to the president of Indianapolis Power & Light, at the Midwest Power Conference held in Chicago earlier this year. . . . In the selections from Mr. Ross' paper which are given here, he tells of the inception of the interchange idea and explains its development since the Lilly contract was negotiated. Although the piping is not described in detail, he includes data about it and in general indicates the application of such contracts and their advantages to industrial plants requiring process steam and nearby utility stations



A view during installation of the 16 in. steam line interconnecting the utility station and the Lilly company

would have lost a customer who is now purchasing large volumes of high pressure steam, generating a portion of its needed kilowatthours in a turbine on its premises, and using the exhaust steam for process work.

Indianapolis Power & Light Co. made studies of the situation at Kingan and Co. which, in view of the experience with the Lilly company, indicated that it would be practical to deliver steam at a pressure of 225 psi with a superheat of 150 deg. It was finally decided that a 2500 kva non-condensing, double automatic extraction turbine generator would operate most satisfactorily. Our analysis indicated that with the customer's process steam requirements going through such a turbine that the customer could generate 10,700,000 kwhr annually which would supply only a part of his electrical requirements, thereby leaving a balance of 5,000,000 kwhr to be purchased directly from the utility. However, under these conditions the power company could not meet the competitive cost of private plant generation, if it used separate billings for steam and electricity.

Combined Steam and Electric Billing

It therefore became necessary to develop a method of combined steam and electric billing which was something new in our experience. It was found that the company could apply its industrial steam rate as well as its large industrial power rate if total electric requirements were billed and a steam credit applied for the kilowatthours generated in the customer's plant. Under this arrangement the steam is billed on the utility's industrial steam rate at an average cost of approximately 32c per 1000 lb. For each kilowatthour generated by the customer he receives credit of 16 lb of steam at the average earned rate for that month. The customer is billed for the kilowatt demand created on the company's distribution system in addition to the kilowatthours generated by the turbine and furnished by the utility. The investment on the customer's premises to adapt his plant to the suggested method of operation amounted to approximately $125,000.

The utility company purchased and installed the turbine generator and accessories at a cost of $75,000, the customer agreeing to pay 12 per cent fixed charges on this part of the power company's investment until such time as he elects to purchase the equipment at its then depreciated value. This arrangement has been working very satisfactorily for both parties. Modifications of the Lilly plan or Kingan plan have been applied to other industries in Indianapolis with the very specific result

to the utility of saving for itself electric revenues which otherwise would not be available because of the existence of isolated power plants.

Steam Headers Tied Together

This installation is served from another high pressure steam line which was constructed between the Perry K and Perry W plants. These two plants, more than half a mile apart, are operated as one with the two steam headers tied together, and the Kingan installation and a similar installation at the plant of the Beveridge Paper Co. enhance the value of this second line from both operating and commercial points of view.

At the present time we have 11 large industrial concerns using so-called high pressure steam, four of whom are generating electricity, but in every case less than their total electrical requirements. A survey of these 11 customers of steam and electricity, among which are a paper mill, brewery, two large hotels, laundry and a federal housing project, besides those mentioned specifically, indicates that whereas these customers formerly provided the Indianapolis Power & Light Co. with some $236,000 annual revenue, they are now providing an annual revenue for steam and electricity of $808,605. This represents a total annual revenue gain of more than half a million dollars, much of which can be attributed to the application of interchange contracts.

INSULATION OF CHILLED WATER PIPING

Jake Stahl, Operating Manager of the Merchandise Mart, Describes Applications of Cork and Wool Felt Insulation

IN my article in the August HPAC, mention was made of the method used in insulating the chilled water piping at the Merchandise Mart, Chicago. On the original air conditioning installations at the Mart, all of the chilled water lines were insulated with cork, and the service was found to be very satisfactory. It was discovered, however, that on some lines removed after 10 years of use, there was considerable rust under the cork. Therefore, on all new installations we are prime coating the pipe before the installation of the cork.

On subsequent installations at the Mart, due to the difficulty of obtaining cork, we have been using wool felt. Double thickness is used with staggered horizontal and vertical joints and the fabric covering is given two coats of paint after installation. Our first installation of wool felt on horizontal lines was not too successful in high humidities prevailing last summer. We found that felt lining was compressed at the hangers to the extent that it lost much of its insulating value and considerable condensation resulted. To some extent, such compression also produces a misalignment of the piping.

Because of its strength in compression, cork insulation is used at the hangers of this 16 in. chilled water piping

With this in mind on the large 16 in. chilled laterals installed this year we decided to use cork at all the hangers because of its strength in compression, with felt insulation between hangers. The cork at the hangers was installed at the time the pipe was hung. Edges of the cork were covered with mastic and then the double felt applied to the length between hangers. A 6 in. canvas lap was used at the joining of the cork and the felt, with mastic applied over the cork and lap strip. We believe that this installation could be improved further by allowing the top lap of the felt to overlap the cork, cutting the cork to receive the lap.

It has been our practice at the Mart to insulate all fan housings with cork, from the dehumidifier chamber to about 10 ft beyond the volume dampers, supplying air to conditioned spaces. Normally we have never had condensation trouble with the ducts beyond this point. We have used duct insulation as such only for the purpose of maintaining the temperatures at the last outlet within a good operating range as compared to the outlets closer to the fan. Usually we have had good success with ducts in furred ceilings without insulation. The one case of condensation which damaged a ceiling was caused between the first and second floors by the large volume of air contained in this 4 ft space and the fact that certain intervening tile walls had holes which allowed considerable air circulation. Patching of these holes stopped the air circulation over the duct and cured the trouble.

RADIANT AND CONVECTION HEATING COMBINED IN GARDEN APARTMENTS

Warm Air Ducts Serve Also as Ceiling Heat Panels . . . . . By G. F. Nightingale

There are eight buildings, each with 16 apartments. The boilers are in one building, from which hot water is piped to all

RADIANT AND CONVECTION heating are combined in the system serving the River Forest garden apartments in suburban Chicago. Hot water from a central boiler installation is piped to a standard unit serving each apartment. From the unit warmed air goes through ducts to ceiling air diffusers. The ducts and portions of the metal ceiling serve as radiant heat surfaces, thus combining these two methods. . . . Mr. Nightingale, western sales manager for Tuttle & Bailey, Inc., describes the layout here, and reports the results of a brief tenant questionnaire

THERE ain't no such animal" as a heating system. Strictly speaking, all systems designed for comfort are in reality cooling systems—or, to put it more scientifically, all such systems are designed to aid the body in ridding itself of its excess heat at a rate which will not put too great a strain on the very efficient thermostatically controlled arrangement which Nature devised to keep us alive (though not at all times comfortable).

In the early days of man's existence on this planet, he learned to keep body heat from being dissipated too rapidly during cold weather by building an open fire in his cave. The bulk of the heat he received by this means came by radiation. For centuries, even after he had learned to build himself a home, radiation was practically his only means of controlling heat loss from his body. It has been only in comparatively modern times that convection, in its numerous forms, has been a factor in controlling body heat loss.

Of late, there has been a lot written about so-called "panel (or radiant) heating," which (in a way) is a reversion to the primitive—since its contribution to comfort is principally through radiation. This is not intended to be a knock against panel heating; in its place, it is an excellent aid to comfort, but only one aid which should, I believe, be used to supplement convection, air motion and controlled humidity.

We have in a suburb of Chicago, a group of apartment buildings in which the architects have employed all four aids, nicely balanced, to condition each space for the maximum comfort of its occupants—the River Forest garden apartments, at Oak St. and Harlem Ave. in River Forest, one of Chicago's "better" suburbs. These apartments were 100 per cent occupied October 1, 1940. There are eight buildings, each with 16 apartments, half of which are 3½ rooms and half 4½ rooms.

Central Boiler Plant

The boilers are in one of the buildings, from which hot water is piped to all. At the boiler house, water temperatures are manually controlled according to outdoor conditions in two ways: (1) the water temperature may be varied by bypassing return water; (2) the volume of water circulated may be varied by adjustment of a variable speed pump, and in exceptionally severe weather an auxiliary pump may be cut in. Although it has not been so operated as yet, the system is designed for use with cool water to provide comfort in summer. Each apartment may be individually

controlled by its separate thermostat; this is extremely important, since few individuals require the same conditions for maximum comfort.

Referring to the floor plan of a typical 4½ room apartment, hot water is piped to a unit situated as shown. This unit is a standard make consisting of a fin type radiator through which air is passed by a standard blower. The air then goes through ducts *A* and *B* to ceiling outlets in the living room and dining alcove and through other ducts to registers in the bedrooms, bathroom and kitchen.

Panel Heat Surfaces

The ceilings have steel veneer surface, attractively painted. Across the living room and the dining alcove, the bottom sides of the ducts form heating panels 9 in. wide by 18 ft for duct *B* and 9 in. by 7 ft 6 in. for duct *A*. These panels are somewhat enlarged by conduction of heat through the steel ceiling. There is also a secondary pair of panels in one bedroom, through the ceiling of which pass the ducts *A* and *B*. Furthermore, some heat from these same ducts passes upward through the concrete slab to the floor of the apartment above (except, of course, on the upper floors). Panel heating is thus provided by these warm air ducts.

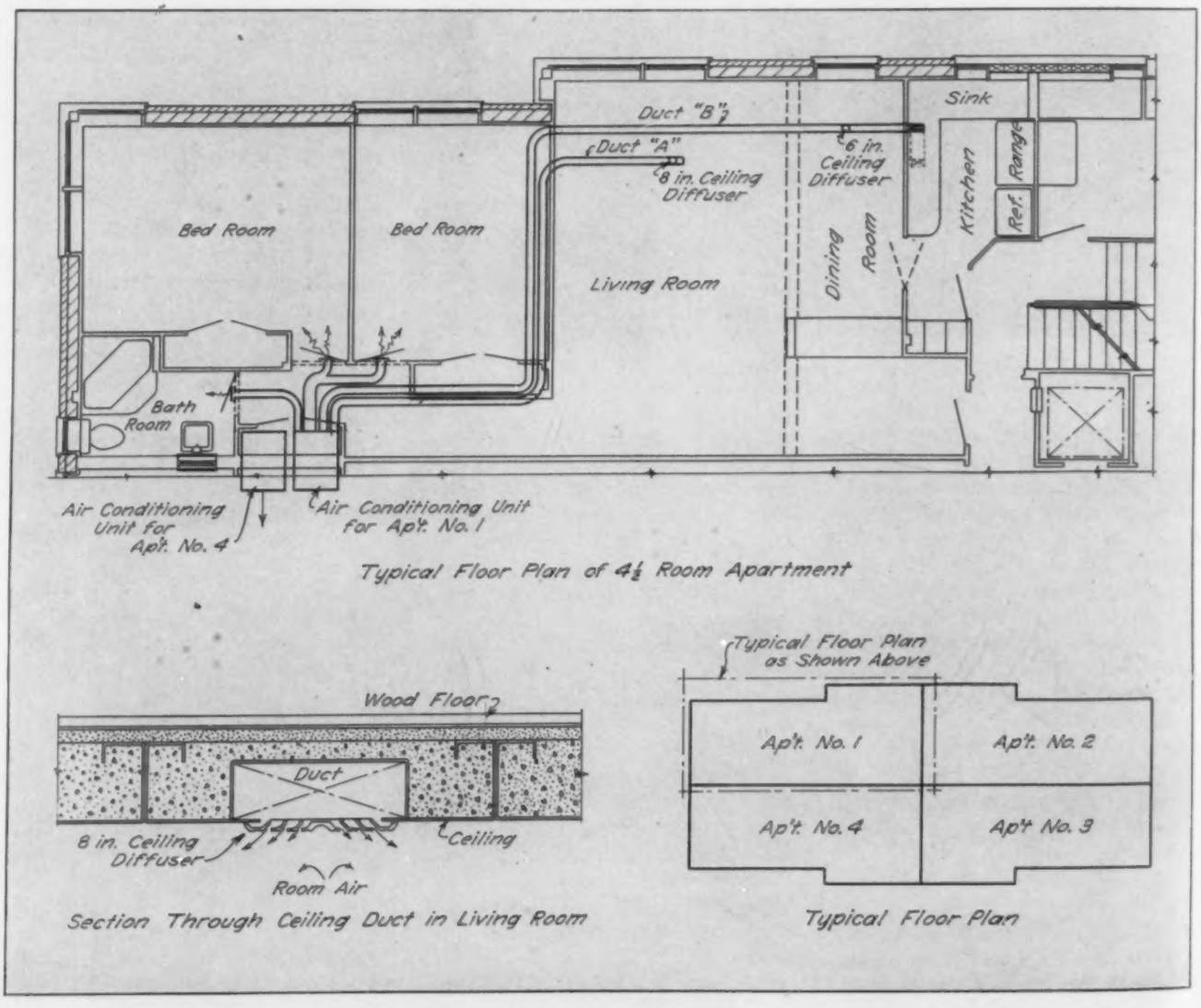
A lot of thought was given to the location of the ceiling outlets themselves. At first, it was felt that a location near the center of the room would give the best results but after careful consideration, it was decided to place both ducts and outlets fairly near the windows, in order to blanket the windows with warm air and also to get the heat radiating surfaces as near the point of maximum heat loss as possible without sacrificing comfort in the back of the room.

Tenant Reactions

A questionnaire was submitted to three families (selected at random) living in the apartment group. Family *X* comprises two adults and two children; family *Y*, two adults and one child; and family *Z*, two adults. Family *X* occupies a northwest apartment.

Note that all three report the same setting of the thermostats in cold or mild weather. This would not be true if the apartments were uncomfortably cool in severe weather. It is natural when we feel cool to try to do something about it and that something is invariably to push up the thermostat. The fact that all three families do not vary the setting during severe weather shows in itself that each is comfortable at the temperature which each selected.

Replies given by family *X* are interesting. Note that their apart-



Living room of one of the apartments (with a peek at the kitchen), showing the ceiling air diffusers

ment has a northwest exposure and that except when the wind is high, they report themselves comfortable. Infiltration through faulty window construction may be—but not necessarily is—the answer.

Caught in the Draft?

In another building, a gentleman without much hair on his head sat under a cool air outlet of the same type used in these apartments and swore he was getting cold from a draft on his head. Tests with smoke showed no downward motion of air; what little air motion there was went up, not down. Probably he did have a feeling of discomfort due to radiation to the diffuser above.

Getting back to family *X*, there is a possibility that the individual who answered the questions might have been unusually sensitive to heat loss by radiation to cold glass surfaces. In that case, like our bald headed friend, she might have thought a draft was causing the discomfort she felt, whereas the true cause might have been an unplanned case of panel cooling not noticeable to less sensitive individuals.

[Architects for the River Forest garden apartments were Holsman and Holsman; heating contractors, Economy Plumbing and Heating Co.; and sheet metal contractors, General Sheet Metal Works.]

Questionnaire Answered by Three Tenants

	X	Y	Z
What is the setting of your thermostat?			
In cold weather	76	72	70-75
In mild weather	76	72	70-75
In cold weather, do you avoid sitting near the windows?	Yes	No	No
Is there any spot in the living room that is uncomfortably cool?	No	No	No
Do you ever experience cold feet?	No	No	No
Do you permit your children to play on the floor?	Yes	Yes	Yes (guests)
Would you classify your apartment as comfortable during the winter?	Except during high winds	Yes	Yes
Do you notice any spot in this room where a draft is noticeable?	No—except near windows	No	No

AIR CONDITIONING AIDS COPPER PRODUCTION FOR DEFENSE NEEDS

In order to increase America's supply of copper urgently needed in national defense work, the Magma Copper Co., Superior, Ariz., has installed three new 140 ton centrifugal refrigeration machines to provide air conditioning down to the 4600 ft levels of its mine. This marks the third extension to the original air conditioning system.

Unusually high rock temperatures have made air conditioning necessary in the Magma mine. Temperatures range from 127 deg on the 3200 ft level to 150 deg on the 4600 ft level. The new installation is expected to cut temperatures down to 90 deg at the deepest workings.

Two centrifugal refrigeration units, supplying cooling equivalent to the melting of 560,000 lb of ice every 24 hr, were installed in 1937. One cooled the air for the 3400 ft level and the other for the 3600 ft level. Each unit is designed to cool 30,000 cfm of air 10 deg.

In November, 1939, a third unit was installed on the 4000 ft level. One of the three new centrifugal machines will be placed on this level, and the other two will be put in operation on the 4200 and 4400 ft levels, the latter machine supplying refrigeration to the 4600 ft depth.

Additions to the present system make the air conditioning system in the Magma the largest mine installation in North or South America. The six units have a total capacity of 840 tons.

ADSORBERS GUARD TELEPHONE EQUIPMENT

In order to combat the adverse effect of common air borne gases upon the proper functioning of automatic telephone equipment, the Bell telephone system is installing adsorbers in the Brandywine telephone exchange at Braddock, Pa. They will extract the gaseous impurities from the ventilating air.

Loss in transmission and amplification efficiency had been traced to the tarnishing of the silver contactors of the automatic selectors. This staining, common to what happens to household silverware, is a film deposit caused by minute quantities of certain gases, particularly sulphur dioxide, entrained in the air coming in contact with the metal.

"Open for Discussion"

Communications to the Editor

AIR DISCHARGE FROM NARROW SLOTS

THE EDITOR—

I have read F. F. Stevenson's article on Air Discharge from Narrow Slots, published in HPAC in May and in June, and I believe it is a contribution to a subject on which very little has appeared. However, I think there are some limitations to the use of some of the formulas that should be pointed out. I have prepared the following discussion based on theoretical considerations and a study of some of the data taken by Professors A. C. Willard and A. P. Kratz in their work on models of ventilating ducts for the Holland vehicular tunnel which may lead some of your readers to a better understanding of the problem under consideration.

The volume of air discharged through a port or slot in the side of a duct carrying air is proportional to the area of the opening and the square root of the static pressure in the duct at the location under consideration. The volume discharged is independent of the velocity of the air in the direction of the axis of the duct when turning vanes are not used.

Since air is removed from the duct, the velocity of the air stream decreases in the direction of flow unless the cross sectional area is proportionately decreased. With decrease in velocity there will be an increase in static pressure because of static pressure regain. However, there will also be a decrease in static pressure because of friction with the walls of the duct. The flow through a uniform slot will increase in the direction of flow in the supply duct if the slot is of such width as to reduce the velocity pressure in the supply duct an amount greater than the friction loss in the section under consideration. The flow through a uniform slot will decrease in the direction of flow in the supply duct if the decrease in velocity pressure and the resultant static pressure regain is less than the friction loss.

In the case of some of the long ducts tested by Professors Willard and Kratz, the static pressure and the resultant quantity of discharge through unit area decreased for a distance from the entrance to the supply duct, reached a minimum, then increased to the dead end of the duct. The decrease in static pressure in the direction of air flow was caused by the high friction loss due to the high velocity of the air in the duct at that end.

This same situation could occur in any long duct with ports or slots in the side. In order to get even distribution of air it would be necessary to have the largest ports at the point of minimum static pressure and decrease the size both ways from this point in proportion to the square roots of the static pressures. In other words, if the static pressure at the end of the supply duct is 50 per cent greater than the static pressure at the point where it is a minimum, the ratio of the width of the slot at that end to width of the slot at the point of minimum static pressure should be 1 divided by the square root of 1.5 or $1/1.22 = 0.82$.

The tests on the models of the ventilating ducts of the Holland tunnel have never been published but they have been referred to in a paper by A. C. Davis published in the Transactions of the American Society of Heating and Ventilating Engineers, Vol. 37, 1931, p. 67. (Also see ASHVE Journal Section, HPAC, October, 1930, p. 870). In this paper Mr. Davis gives the following formula developed by the late Professor Goodenough for calculating the total pressure at any location in a duct supplying air through ports or slots in the sides:

$$T.P. = \frac{12y}{D}\left\{\frac{W_1^2}{2g}\left[\frac{aLZ^2}{3m} - \frac{1}{2}(1-K)Z^2\right] + \frac{bLZ}{2gm^2}\right\} + P_o$$

where $T.P.$ = total pressure in inches of water;

y = density of air in duct, lb per cu ft;

D = density of water, lb per cu ft;

W_1 = velocity of air entering duct, fps;

g = 32.2 ft per sec per sec;

a = 0.0035, a constant;

b = 0.01433, a constant;

L = total length of the duct, ft;

$Z = \dfrac{L-X}{L}$;

X = distance from the entrance of duct to any chosen section;

m = hydraulic radius of the duct in ft = area of the duct in square feet divided by perimeter in feet;

K = 0.615, a factor allowing for turbulence in the duct;

P_o = bulkhead pressure in inches of water (or the static pressure at the dead end of the supply duct).

This formula could be used in the solution of a practical problem by calculating the velocity pressure corresponding to the desired velocity of exit from the slot or port and taking that pressure as the bulkhead

Readers of HEATING, PIPING AND AIR CONDITIONING are invited to contribute their views to these "Open for Discussion" pages, which will appear from time to time. Comments on articles we have published, expressions of timely interest on developments in design, installation, operation or maintenance of heating, piping or air conditioning systems, other remarks of value to HPAC's readers — all will be welcomed by . . . . THE EDITOR.

"Open for Discussion"

pressure P_0. The total pressure which the fan must provide at the entrance to the supply duct may be calculated with the formula by taking $X = 0$ in which case $Z = 1$. The static pressure at any distance X from the entrance, may be calculated by subtracting the velocity pressure in the supply duct, given by the formula

$$V.P. = \frac{12y}{D}\frac{\left(\frac{L-X}{L}W_1\right)^2}{2g},$$

from the total pressure calculated for the same distance from the entrance.

A number of values for X between 0 and L may be taken and the static pressure calculated and plotted as ordinate against the distance from the entrance X, as abscissa. The static pressures at points between the locations for which the static pressures were calculated could be read from the curve.

Friction Loss Formula

The formula given on p. 310 of Mr. Stevenson's article in the May HPAC for finding friction loss appears to be based on the assumptions that the friction loss varies as the square of the velocity and that the velocity at the dead end of the supply duct is zero. Both assumptions are correct and the formula should give an approximate value for the friction loss for the entire length of supply duct but could not be applied to the calculation of friction loss from the entrance to any point between the entrance and the end because the friction loss per foot of duct is not the same for different portions of the duct. For example, if we were considering the friction loss from the entrance to the midpoint, the "empirical constant" in Mr. Stevenson's formula should be about 0.56 and if we were considering the friction loss between the midpoint and the dead end the constant would be about 0.062. In any case, the use of Mr. Stevenson's formula would give only an approximation of the correct value.

Static Pressure Regain

In calculating static pressure regain in the supply duct, the fact should not be overlooked that air is removed from the duct and that the mass of air flowing continually decreases. Professor Goodenough took this factor into consideration in deriving the formula given in Mr. Davis' paper.

I believe that if all slots or ports are properly proportioned to each other and the supply duct, that even distribution of the air will ensue regardless of the velocity of the air entering the supply duct. If the discharge from ports near the entrance is too small, it is because the static pressure is too low, not because the velocity is too high.

I am unable to discover any rational basis for the method given by Mr. Stevenson on p. 370 of his June article for calculating the throw of air from a slot, and would not recommend its use without a thorough explanation of its derivation or complete experimental proof of its validity.—J. R. Fellows, assistant professor of mechanical engineering, University of Illinois.

COMPLEX PROBLEM

The Editor—

The movement of air through a duct and discharge from slots offers a complex problem with many variables. Present research in various laboratories is adding to our knowledge of the subject and will make it easier for the designer to obtain a satisfactory solution. It seems impossible and inadvisable to attempt to take into account all of the variables, so as a necessary part of a duct system there will be included adjusting or balancing devices. This is true whether slots or some other form of outlets are used. An adjustment of the width of slots would seem to be very important. Actual installations in their period of adjustment often can serve as laboratory installations for the furtherance of knowledge.

It is stated by F. F. Stevenson in his article in the May HPAC that the field for slots is not in comfort air conditioning but in industrial applications. Essentially there is very little difference between a slot and a series of long, narrow grilles. An open hole is a poor grille and a slot only works because of the high ratio of length to width. To my mind a duct system with a succession of grilles could be designed to function satisfactorily without much added cost over a system using slots.

Diffusing Grilles

Grille manufacturers have elaborate tables based on laboratory tests to show the performance of their products. Many variables affect this performance, but the designer need not use complex formulas to predetermine performance. Similar tables could be determined for open holes which in one extreme of aspect ratios become slots. However, tests seem to show that the performance of diffusing grilles is so much better that the application for open holes would seem very limited. Where ceilings are high and where throws are short, slots are workable. But in ease of selection and in control of essential factors such as noise, tilt of air stream, length of throw and entrainment of room air, the diffusing grille seems superior. When the cost of the entire heating and ventilating or air conditioning system is considered, how much would the use of grilles add to the cost over that for the use of slots? And if the costs are compared on the basis of the total building costs, the difference would seem negligible. Further than that, the building is usually used to house expensive equipment with which production is the important item. Any small initial increase in investment due to grilles is likely to earn dividends. Why put in fans, heating coils, cooling coils, humidifiers, control apparatus and omit diffusing outlets?

Aspect ratio has an effect on the performance of openings but it is usually unimportant below a ratio of about seven. With slots it undoubtedly is important, since the aspect ratio approaches infinity. The greater the aspect ratio, the shorter the throw due to the larger amount of contact surface between the jet and the room air being put into circulation. There are many additional factors influencing the discharge

"Open for Discussion"

from a given arrangement of duct and opening.

Discharge Coefficients

Slots are distant relatives of orifices and nozzles, which have widely varying discharge coefficients ranging from about 0.6 to 1.0 depending upon the degree of guidance exerted on the air stream. The height and contour of a slot affect the discharge coefficient and if a throat or neck were used, the coefficient would likely be higher. The range from 0.55 to 0.85, as mentioned by Mr. Stevenson, appears reasonable.

The statement is made by Mr. Stevenson that installations with slots were observed where, due to the spring of metal, the jet was slowed up by reaching the ceiling, with a consequent shortening of throw. The throw from slots is short due to large entrainment surface and if the air stream slides on the ceiling the throw will be increased. Only where the angle of hitting is such as to cause a bounce, and therefore a drop in the air stream, will the throw be shortened.

The laws of fluid mechanics seem to state that the flow from a side wall opening in a duct depends only upon the static pressure in the duct and not at all upon the velocity pressure in the axial direction. This principle is made use of in the usual static pressure device, small holes about 0.04 in. in diameter through the duct wall connected to a slope gage. When such holes are enlarged to, say, ½ in. diameter some velocity effect is measured. It seems logical when the hole becomes infinite in length, as is the case with a slot, that the velocity effect would be appreciable. Likely the air leaves a plain slot with an angular discharge such that the perpendicular component depends upon the square root of the static pressure and the component parallel to the duct will be proportional to the square root of the velocity pressure. In actual cases even in laboratory set ups, many factors, such as turbulence and slot edge effects, disturb this relation. However, there is no reason why air escaping at a given section should not retain most of the velocity energy it had within the duct until such is lost in putting the room air into motion. This applies to the case without turning vanes. With vanes the duct velocity is made available for motion perpendicular to the duct.

At a given duct static pressure and velocity, a turning core or vanes would increase the velocity of the air stream leaving the duct slot. The vanes should reduce the pressure drop through the slot discharge due to decreasing turbulence in spite of added friction surface. Tests on elbows used as stack heads show a very great reduction in static pressure at the base of the elbow due to vanes.

Static Regain

The principle of static regain is well known and is frequently used in all types of duct distributing systems to maintain static pressure against friction losses. A most complete outline and charts for this method of designing ducts is described in the book Modern Air Conditioning, Heating and Ventilating by Carrier, Cherne and Grant.

Throw of Air

The method for determining throw given by Mr. Stevenson on p. 370 in June appears to be over-simplified and lacking a logical basis. It might be likened to the formula used at times to determine the taxable horsepower of automobile motors in some states, where so many variables have been omitted from the basic formula or made constant that the formula is grossly in error. Unfortunately, it is not easy to state the correct formula for throw. Some of the variables omitted in the method given are discharge coefficient, aspect ratio, distance to floor and ceiling and temperature difference.—D. W. Nelson, associate professor of mechanical engineering, University of Wisconsin.

WHY PANEL HEATING?

The Editor—

Having noted with interest how certain recent letters on this page have drawn fire from antagonist and protagonist, I ask the privilege of tossing in a question regarding that type of heating which is currently getting much publicity and which in its various forms is known as radiant heating or panel heating.

One reads the most scientific analyses of the principles of radiant heating by some of our most able physicists, and one sees impressive photographs of installations in buildings bearing the names of the country's leading architects; but does not this question arise in the mind of others as it does in mine—*why?*

Why is it that after years of evolution and improvement of convection or semi-convection heating, after great effort to perfect methods of control, and after having arrived at reasonably satisfactory methods of calculation, are we suddenly presented with the radiant heating method—with the implication that it is the perfect method, or at least better than those we have used?

It would be appropriate for someone to present a case for radiant heating, demonstrating whatever real advantages it may have before too much effort is wasted upon its study.—J. H. Walker, superintendent of central heating, The Detroit Edison Co.

FUEL OIL

The Editor—

This will acknowledge your letter, together with the enclosed articles written by Kalman Steiner, on the fuel oil situation, which are very interesting. In the July article mention is made of inland waterways and the possibility of utilizing motor tank truck transportation on the highways; this is particularly pertinent because of the fact that we have already taken the position that if the highways can be used to offset the pressure on railroad tank car and pipe line transportation, they should be given every opportunity to be utilized.—H. A. Gilbert, director of transportation, Office of Petroleum Coördinator for National Defense.

CHILLED WATER CIRCULATING SYSTEMS FOR STORAGE AIR CONDITIONING JOBS

Bill Goodman Tells How the Piping Should be Arranged, Gives Pointers on Proper Control

STORAGE TANKS—Where chilled water or brine is circulated for air conditioning or refrigerating purposes, the use of a storage tank permits operating economies to be achieved. By using a storage tank, a reciprocating compressor can be operated at practically full load whenever it is running. Under light load, the running time of the compressor will be about directly proportional to the cooling load with resultant economy in electricity and condensing water. Furthermore, the use of a storage tank prevents short cycling of the compressor under light load. . . . The method of properly sizing storage tanks so that they are neither too large nor too small was discussed last month by William Goodman, consulting engineer, The Trane Co., and member of HPAC's board of consulting and contributing editors. This month, he describes open and closed circulating systems, proper position of the balancing tank, venting, etc. He also presents recommendations for controls and stresses the importance of thorough mixing of the returning water in the storage tank

TWO types of piping systems are commonly used for circulating chilled water or brine from storage tanks to the air cooling coils or heat transfer surface. The *closed* system is illustrated in Fig. 1 and the *open* system is shown in Fig. 2. In the open system of Fig. 2, the water cooling coils are immersed in the storage tank. Installing cooling coils in the storage tank, as in Fig. 2, results in a somewhat lower first cost. However, where a separate water chiller must be used, a closed system of the type illustrated in Fig. 1 is desirable because a single pump can be used for circulating the water between the water chiller, the storage tank and the air cooling coils.

In both open and closed systems, if the air cooling coils are placed at the high point of the system, the water supply pipe should be connected into the bottom of the air cooling coil, and the water outlet pipe at the top, as illustrated in both Figs. 1 and 2. In this way, when the system is filled, the water will sweep the air ahead and out through the vent in the case of the closed system and through the open end of the outlet pipe in the open system.

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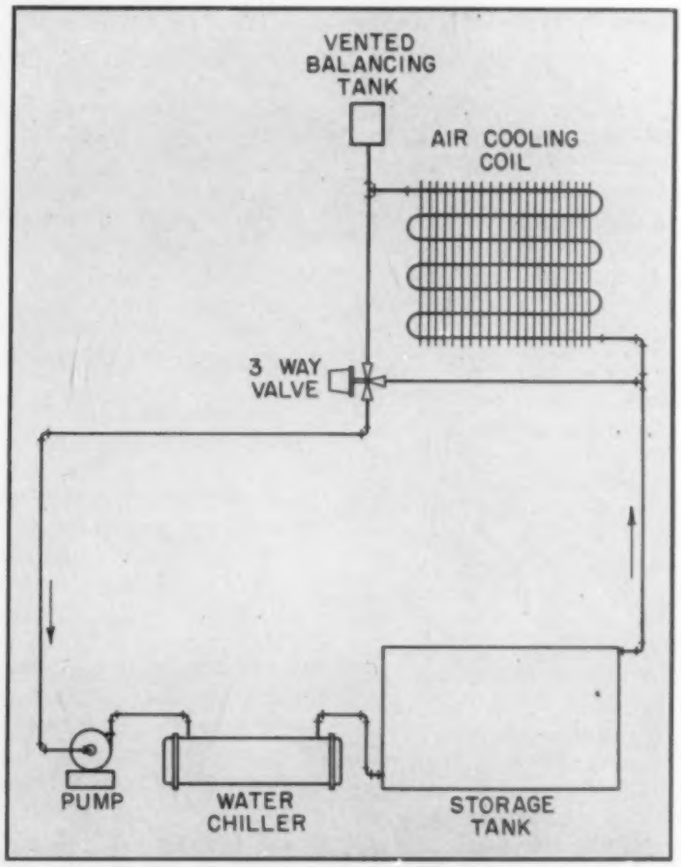


Fig. 1—Closed circulating system

Balancing Tank

In closed systems, the location of the balancing tank is of importance if a vacuum is to be avoided in the circulating piping. A vacuum in any portion of a closed circulating system is undesirable as air tends to pull in through every slight leak. The air may accumulate at various points in a closed system where it cannot be vented conveniently. If sufficient air accumulates in any part of the system, difficulties may be experienced with the water circulation.

By placing the balancing tank on the suction line to the pump, as in Fig. 1, the gage pressure at the pump suction will always exceed atmospheric pressure by the dif-

ference between the gravity head on the pump and the friction loss in the piping between the balancing tank and the pump.

In the closed system of Fig. 1, if the balancing tank were placed at the inlet connection of the air cooling coil, instead of at the outlet as shown, the entire cooling coil might be under a slight vacuum. The pressure at the inlet to the coil would be atmospheric, but due to the friction loss in the coil, the pressure of the water inside the coil would fall steadily. The pressure would become equal to atmospheric pressure again somewhere in the return piping where the gravity head was sufficiently large to offset the friction loss. In this case, any air that leaked in through the connections around the coil would either be trapped at the high point of the coil or would be carried through the piping and the pump, and might be trapped somewhere else in the system—possibly in the water storage tank. For this reason, the balancing tank or venting device should always be placed at the outlet of the coil and on the suction line to the pump as shown in Fig. 1.

Avoiding a Vacuum

If all the equipment in a closed circulating system is — practically speaking — situated on the same level, it is especially important correctly to locate the balancing tank, and any additional venting devices needed, to be sure that a vacuum does not occur at any point in the system. In this case no static head is available to assure positive pressure in the system. If the balancing tank were placed at the inlet to the air cooling coil of Fig. 1, and if the static head were negligible, the entire coil and all the suction pipe between the coil and the intake of the pump would be under vacuum. Where the suction piping between the coil and the pump is very short and a slight static head exists—which is frequently the case—the balancing tank can be situated as in Fig. 1.

However, if there is a long run of suction piping between the air cooling coil and the pump, and the static head is negligible, the entire suction line may be under vacuum if the balancing tank is at the coil outlet, as it is in Fig. 1. In this case, to avoid a vacuum in the suction line and at the same time to allow the air to vent from the high point of the system—which is usually at the air cooling coil—a separate closed type float vent can be used at the outlet of the air cooling coil and the balancing tank can be connected directly into the suction of the pump as illustrated in Fig. 3. In this case, situating the balancing tank at the suction of the pump insures that the lowest pressure in the system at the pump suction will be approximately equal to atmospheric pressure. The pressure throughout the piping system and the air cooling coils will be greater than atmospheric in the system illustrated in Fig. 3. The float vent at the high point of the system will also be under pressure. When sufficient air has gathered in the float chamber, the float valve will open, discharging air to the atmosphere—after which the float will again close. The air discharge from the float vent should be so piped that the air is discharged above the drain pan of the air cooling coils.

Disadvantage of Open System

The disadvantage of the open system illustrated in Fig. 2 is that the pump must discharge the water not only against the friction head of the entire system, but also against the gravity head—that is, the vertical distance from the water level in the storage tank to the high point of the coil. A coil submerged in the storage tank (as in Fig. 2) can also be used in a closed system. In this case, a vented balancing tank must be provided on the outlet side of the air cooling coil connected in the manner illustrated in Fig. 1.

The disadvantage of pumping against the entire gravity head in an open system can be almost entirely offset, if the vertical elevation is not too great, by so designing the open system that a vacuum is maintained in the air cooling coil. In the open type system of Fig. 2, there is no disadvantage in having the air cooling coil—or any part of the system which is situated between the discharge of the pump and the open outlet of the discharge pipe—under vacuum. Any slight amount of air that leaks in will be vented continuously from the system through the open discharge pipe. In this case the head on the pump will be reduced by the amount of vacuum maintained at the high point of the system. The method of obtaining and using a vacuum in order to reduce the pumping head on an open system when circulating cold water has been discussed in HPAC, July and August, 1937, pp. 418 and 477, respectively.

Using System for Heating

In an open system, if the same coils and circulating water are to be used in winter for heating the air, it is better to install a pump capable of discharging against the full static head of the system and maintaining atmospheric pressure at the high point of the coil. If a high vacuum is maintained inside the coil, some warm water may flash into steam. Utilizing a vacuum at the high point of an open system in order to reduce the discharge head on the pump is feasible only when cold water is to be circulated through the coil.

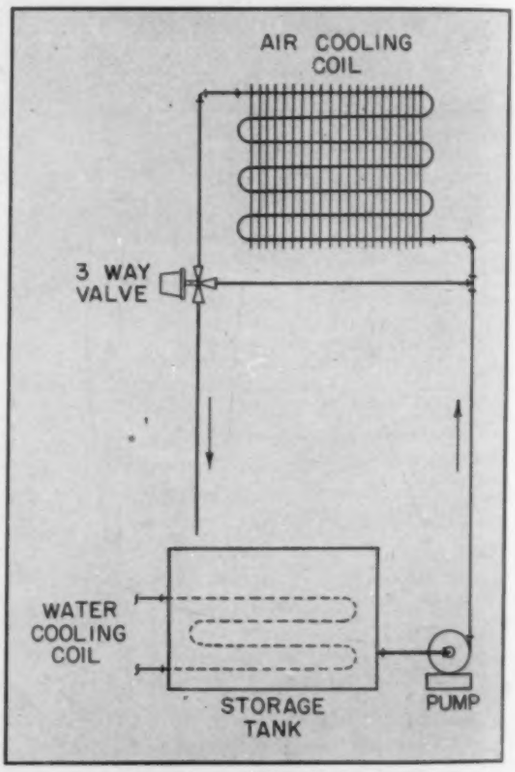


Fig. 2—Open circulating system

Controls

The quantity of chilled water supplied to the air cooling coils should be controlled by means of a three-way valve, as illustrated in Figs. 1 and 2. By using a three-way valve, only the quantity of water actually required for the cooling load will be admitted to the air cooling coil. The excess chilled water will be bypassed back to the water chiller and the storage tank, thus further cooling the water stored in the tank.

If a number of separate air cooling coils are to be used in a circulating system, separate three-way valves can be provided for each coil or group of coils as desired, each three-way valve being actuated by its own thermostat.

In a circulating system with a separate water chiller as illustrated in Figs. 1 and 3, running of the compressor should be prevented unless the water circulating pump is also running. This can be accomplished by so interconnecting the electric wiring that the compressor cannot start unless the pump is also running. Better even than an electric interconnection is the use of an orifice with a differential pressure electric switch in the discharge pipe of the pump. With such a differential pressure electric switch, even though current is on the motor, the compressor will not be able to start unless sufficient water is being circulated so that there is no danger of a freeze-up. The differential pressure switch can be so adjusted that it will not start the compressor unless the minimum desired quantity of water is flowing.

Mix Stored Water Thoroughly

Storage tanks should be so designed that the water returning to the tank is thoroughly mixed with the water already in the tank. Short circuiting of the water entering and leaving the tank can be avoided by properly situating the inlet and outlet connections on the tank. If necessary, a partition should be installed inside the tank to assure thorough mixing.

The thermostat starting and stopping the compressor should be placed in the tank at a point where it is certain to be actuated only by the temperature of the water mixture. It should be in a position remote from the water inlet connection.

By using the data on pp. 502-505 of the August, 1941, HPAC storage tanks can be sized intelligently so that they are neither too large nor too small for the intended purpose. In addition, by using balancing tanks and vents as illustrated in Figs. 1 and 3, difficulties with air binding can be eliminated.

[In the August HPAC, Mr. Goodman explained a method of sizing chilled water (or brine) storage tanks, and presented tables showing on and off times of compressors, and water storage required in gallons per ton of compressor capacity.]

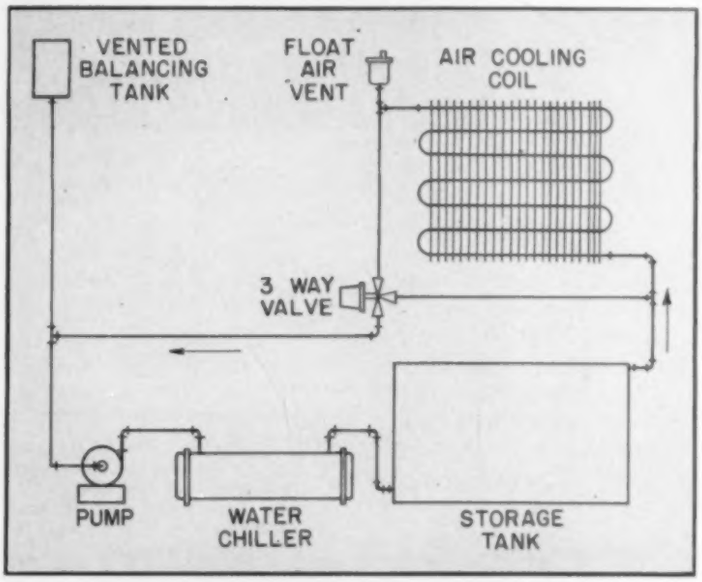


Fig. 3—Closed circulating system in which all equipment is practically on same level; that is, gravity head on the pump is negligible

SEEKS STANDARDIZATION OF VALVES ON NITROGEN BOTTLES

The G & N Mfg. Co., Cleveland, maker of die casting machinery, has launched a campaign for mandatory standardization of valves used on industrial nitrogen bottles.

This action grew out of an explosion in a die casting plant which killed three men and injured a fourth. The explosion occurred after the accumulator tank on a die casting machine had accidentally been filled with oxygen instead of the nitrogen which had been intended for it. The oxygen came into contact, under pressure, with hydraulic fluid used in operation of the machine, and a chemical reaction was set up which caused the oil to ignite and burn at tremendous speed.

G & N seeks to have the National Bureau of Standards introduce a regulation making it necessary for nitrogen producers to sell nitrogen only in bottles with left-hand threads on the valve fittings. If this is done all nitrogen inlets will have corresponding threads and it will be impossible to attach them to an oxygen bottle.

The danger of explosion through the contact of oxygen and oil is not limited to the die casting industry. In the utility industry, for instance, both oil and nitrogen are used in contact with each other in certain types of generators, and were oxygen accidentally substituted for nitrogen an explosion would occur, according to G & N.

Similarly, some oil filled power transmission cables are now being emptied of oil and filled with nitrogen, and here again if oxygen were substituted the oil would explode with disastrous results, G & N states.

Ignorance Is No Excuse

. . . . THE LAW PRESUMES INDUSTRY TO HAVE KNOWLEDGE OF ITS HEALTH HAZARDS, TAKE MEANS TO PREVENT THEM, SAYS LAWRENCE R. BLOOMENTHAL IN THIS REVIEW OF THE SUBJECT

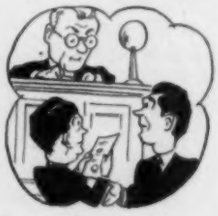
AS industry speeds up to meet the demands of the defense program, it is imperative that safety devices already in use be rechecked to determine their adequacy. With prolonged hours of operation and a constant influx of new workers, industrial health hazards are likely to increase, particularly in those plants where gases, fumes and dust result from the processes of manufacture.

Lack of Knowledge

Directly in point is a recent decision which declares that an employee is not presumed to possess the scientific knowledge necessary to appreciate the dangers involved in handling certain types of material. A workman who had operated an emery grinding disk for smoothing down leaded joints sued his former employer, alleging that lead poisoning had been contracted because neither a mask nor proper suction apparatus had been furnished.

This charge of negligence was defended by the manufacturer, who contended that the risk of poisoning was an obvious accompaniment to the occupation and one which was assumed voluntarily by accepting employment. The court's decision, however, made it clear that there was no justification for this reasoning and the defendant was forced to pay damages for failure to provide preventive devices.

It is significant that an employer is presumed by law to have a greater awareness of occupational risks than his employees. This theory was invoked in a case where the owners of a glass factory were charged with knowledge of dangerous dusts in their plant. The evidence showed that the sole means of ventilation consisted of open windows and a dependence upon the natural forces of gravity. It was established by expert testimony that in other similar plants, respirators and ventilating facilities were in common use to clear the air of soda ash and other dusts.



INDUSTRIAL HEALTH HAZARDS — With prolonged hours of operation and a constant influx of new workers, due to the demands of the defense program and the general rise in business activity, industrial health hazards are likely to increase, particularly in those plants where gases, fumes and dust result from manufacturing processes. Mr. Bloomenthal, an attorney, cites a number of specific cases which have come up in various states in this brief review of the subject. His selection of the material brings out a number of the legal principles involved

The defense maintained that the employer was unaware of the presence and peril of this material and consequently could not have foreseen the particular injury. In reply, the opposition brought out the fact that a company physician was in regular attendance at the factory who knew, or could have known, of the harmful effect of soda ash. In sustaining a verdict for substantial damages, the court stated that even if it were true that knowledge of actual conditions was lacking, failure to comply with prevailing safety standards established inexcusable neglect.

Inexperienced Employees

Merely warning inexperienced help that the work may expose them to the inhalation of poisonous or irritating fumes, dusts or gases is not enough. The courts insist, even in the absence of statutory regulations, that there is the further duty to supply goggles, respirators or other protective equipment.

In a middlewestern state, a farm hand was taken on as a helper on a road construction job. He was put to work in the area between the concrete mixer and a platform where cement, sand and gravel were unloaded from supply trucks. A continuous cloud of dust was present throughout the working day but no instructions were given as to the necessity for wearing a mask, nor was any other protection afforded. After working about five months, the employee became ill from a lung ailment which physicians later testified was incurable.

Both the foreman and the superintendent of the construction company stated that goggles and respirators were kept on hand at the mixer, and had they been worn it would have been safer for the helper. They admitted, however that no one had ever offered them to him. Further evidence proved that there was a general custom for contractors in this business to provide such devices where these conditions prevailed.

In addition to an award of $1000 for eye injuries the jury granted $6000 damages to the plaintiff for the lung disease resulting from cement dust. The fact that the doctors could not agree upon a definite classification for the specific type of lung trouble was held not to affect the right to recovery. Since respirators were actually on hand it was apparent that the company knew the dangerous nature of the work and should have taken positive steps to guard against it.

Location of the Work

One of the arguments in the cement dust case was that the location of the work—the open air—made it unnecessary to take special precautions. This same contention was raised by a corporation manufacturing artificial silk thread, cellulose, alcohol and other products requiring the use of sodium carbonate. Under

the laws of the state where the plant is situated, exhaust fans or other means to purify the air are compulsory in any establishment where noxious fumes are liable to be inhaled by the workmen.

Claiming that he had contracted a form of tuberculosis from exposure to soda ash dust, an employee sued the company for a large sum. The corporation attempted to refute the charge by stating that this particular job was performed outdoors, so that the above-mentioned law did not apply. According to the testimony, however, while the plaintiff's work was outside, it brought him in close proximity to the exhaust end of a pipe connected with the dust collecting system in the plant.

Upon an appeal from an adverse judgment in the lower courts, the defendant took the case before the highest court in the state. Here also the decision went against the employer. In its opinion, the court held that the location of work did not necessarily lessen the obligation to provide a safe place of employment. Whenever harmful dusts are prevalent, proper appliances are essential whether the work is performed within or without a building.

Modern Equipment

The fact that an employee has sustained injuries does not in itself establish conclusively that working conditions were unsafe. Whenever suit is filed to recover for occupational illnesses, excellent defense is clear proof that the most modern equipment has been installed and is being maintained efficiently.

One such case arose when an ex-employee sued the owner of a sand grading and pulverizing plant, declaring he had contracted silicosis from the inhalation of sand particles. His work included the operation of a compressed air gun to prevent the clogging of a vibrating screen. It was admitted that there was no ventilating system at this point in the screen tower, except windows which had been ordered closed.

However, detailed testimony revealed that before the erection of the plant, the president of the corporation had traveled extensively, accompanied by engineers, in order to study conditions in other dusty industries. All the equipment was planned on the basis of this research and was constructed by large and experienced concerns. Expert witnesses testified that the system installed was the best then available and even a little more advanced than prevailing standards. Furthermore, it had been approved by the state industrial commission having jurisdiction.

The men were all provided with sponge masks which they were warned to use. Several experts agreed, at the time of the trial in 1932, that these were preferable to "air line" masks which were still in an experimental stage. Holding that an employer is not an insurer of the absolute safety of his employees, the court decided that since every reasonable precaution had been taken, there was no liability for injuries. The charge that windows should have remained open was considered also, but the plaintiff was unable to offer substantial proof that this would have made enough difference to have prevented contraction of the disease.

Statutory Violations

While remedies under common law have long been available, many states have enacted statutes to compel employers to provide means for removing dust and fumes. At common law, the employer is duty bound to furnish a safe place in which to work and to supply whatever equipment or devices may be necessary for protecting the workmen. One important test of compliance is whether the means adopted conform to those generally approved and customarily used in the industry.

A decision of the supreme court of one state, handed down in 1939, illustrates the gradual trend toward the imposition of stricter obligations upon manufacturers. The statute in force at the time of the injuries directed that exhaust fans or other devices of sufficient power be maintained for carrying off poisonous fumes, gases and dusts from emery wheels, grindstones and other such machinery. Until proper safety measures were taken, use of this machinery could be prohibited by the chief factory inspector.

The plaintiff in the case charged that her deceased husband had been employed for 11 years as the operator of a "segment grinder" used for polishing ceramic or pottery products. His death occurred within seven months after admittance to a tuberculosis sanitarium which he entered directly after leaving his job. It was alleged that no proper dust collection hood had been provided, that there was no sufficient suction device for carrying away the dust and that the defendant was negligent in not supplying a mask of the customary type.

As a defense, the pottery company proved that it had installed a dust removing appliance similar to that generally approved and then in use. Other defenses were that the employee had undertaken the work with full knowledge of the risk and had been contributorily negligent in not wearing a mask which was on the premises. However, the jury was not convinced by the evidence on this latter point.

In rebuttal, testimony was introduced which demonstrated that not more than 25 per cent of the dust generated by the grinding process was eliminated by the dust collecting device and that it frequently was out of repair. A judgment for the plaintiff was affirmed on the ground that merely installing an approved system does not in itself fulfill the employer's statutory duty. It must be inspected and kept in good working order; otherwise, it is not sufficient within the meaning of the law. Permitting safeguards to become inefficient nullifies whatever benefits might be derived from the installation of the equipment.

There is a statute in a middle-western state which provides that every employer in that state whose business subjects employees to illness or disease shall provide approved and effective means to prevent occupational diseases. Additional precautions are ordered for any process, manufacture or labor involving antimony, arsenic, brass, copper, lead, mercury and other specified chemicals or their acids and fumes.

All factories within this classification are required to maintain adequate and approved respirators and to cover all hoppers, chutes and

similar machinery with a hood. Based on alleged violations of this statute, suit for $15,000 damages was filed by a workman previously employed as a laborer in a paint plant. His duties included the carrying of dry lead, paint and whitening from one section of the shop to another.

Part of his work took him into the mixing room and he claimed to have become afflicted with "painter's colic" from inhaling large quantities of dust and vapors. The complaint set out the various aspects of negligence, including failure to provide a reasonably safe place in which to work, absence of proper ventilators and respirators, lack of covers for the mixing machines and insufficient warning despite the employer's comprehension of the poisonous nature of the fumes.

The employer corporation argued that the statute was unconstitutional because it was too vague and uncertain. It was claimed that the standards prescribed were too indefinite, since they did not designate the kind and character of devices to be furnished by the employer.

The supreme court of the state upheld this act, stating that in view of the benefits to be derived by protecting public health, it was unnecessary for the state legislature to outline the exact methods to be followed. An approved device means one which has been publicly recognized as suitable for preventing injuries against which the law was aimed. Furthermore, the corporation could not complain that it did not know what course of action would satisfy the law, since the evidence showed that no safety apparatus whatsoever had been installed!

Gas Masks

As recently as 1939, the courts of another state sustained the constitutionality of a statute requiring "serviceable gas masks" to be supplied to employees working in an enclosed room in which dangerous, noxious or deleterious gases are present. In a coke processing plant, the oven gas was conserved, reduced to crude benzol and then refined into a salable product by adding sulphuric acid and mechanically agitating the mixture.

While draining off impurities from the bottom of the agitating tanks through an open trough into another structure, a workman was overcome several times by the fumes from escaping benzol gas. Despite his protests, he was not furnished with a gas mask nor were any mechanical changes made to alleviate the condition. Finally, because of serious disabilities, he was forced to stop work altogether and filed suit for damages.

The company based its defense on the ground that the statute was too ambiguous because the phrase "serviceable gas mask" had not been defined. Overruling this objection, the state supreme court held that statutes prescribing safety and sanitary regulations need not be drafted with meticulous precision. Since they are intended for protection of workers in industry, they are valid if they clearly designate the hazards and reasonably indicate the means by which protection is to be accomplished.

In reviewing the evidence, the court stated that not only were the masks kept in a separate building some distance away but that the work could have been made less dangerous by means of fans and glass enclosures. These changes would not have impaired the efficiency of the plant and would have eliminated reliance upon open windows and vents for ventilation purposes.

GAS ENGINE DRIVES COMPRESSOR

According to Porter Morgan, manager of the Hilton Hotel, Longview, Texas, the engine illustrated operates on natural gas and replaces the former drive for a 50 ton compressor for air conditioning 65 of the hotel rooms, the coffee shop, and the dining room. The cost of the gas required is approximately $1.00 per day. During the month of July—which was a hot one, with temperatures ranging most of the time from 95 to 100 F—the gas engine burned 383,800 cu ft of gas at a cost of 10c per thousand. Oil is changed approximately twice a month or around each 400 operating hours, cost of each oil change being around $5.00. . . . Savings in one season almost pay for the new equipment and no additional personnel is required for operation. The engine is equipped with an automatic oil pressure cut-out and high temperature control, and has a recording clock showing hours of operation. It is cooled by a closed system using an outside tower.

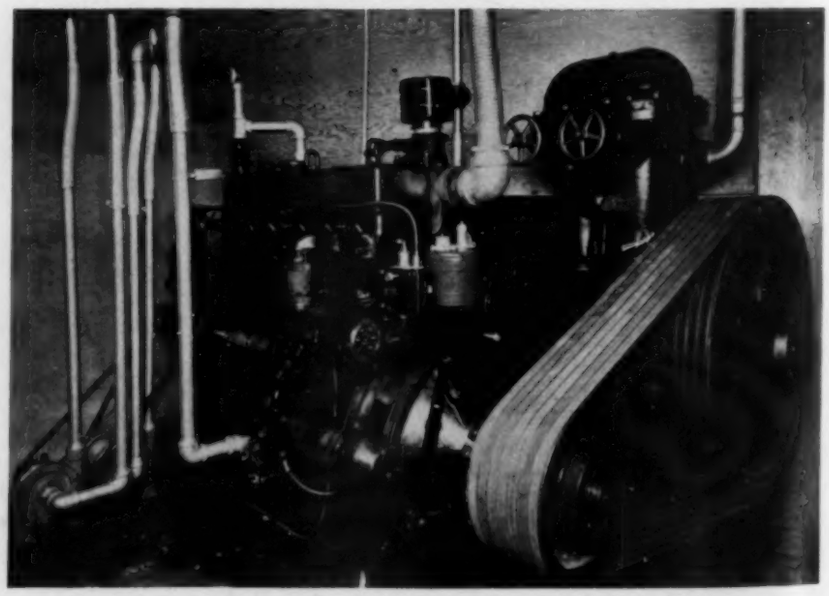


Photo courtesy of Continental Motors Corp.

Production for National Defense

GAGES ESSENTIAL TO NATIONAL DEFENSE PRODUCTION; AIR CONDITIONING ESSENTIAL TO GAGE MANUFACTURE

A. S. MacNeish Describes New Gage Building of the Greenfield Tap & Die Corporation

THE new gage building of the Greenfield Tap & Die Corp., at Greenfield, Mass., is a two story and part basement structure, approximately 200 ft long by 80 ft wide, constructed especially for the finishing operations and inspection of precision built gages. The building is of brick and concrete construction, has very few single glazed windows but has large areas of glass block surfaces in the north and east walls. The roof is insulated and flooded. Fluorescent lighting is used throughout the building, maintaining a uniform average intensity of 22 footcandles.

To facilitate and improve on the manufacture of this equipment, made regularly with tolerances of 1/10,000 in., the building is air conditioned throughout. The air conditioning installation consists of two distinct systems, one serving the manufacturing space on the two floors of the building, called the "building system" and the other serving the final inspection room.

The building air conditioning system evolved from various requirements of the owners, viz: mechanical ventilation, which requires indirect heating; elimination of atmospheric and manufacturing dust, requiring filters; and stabilizing of temperatures, which requires heating and cooling.

It should be noted that stabilizing of temperature—and not comfort conditioning, a secondary consideration—was required. The purpose of this was to eliminate hour to hour temperature fluctuations, which cause continuous expansion and contraction and which would do no appreciable harm if the measuring devices and the work expanded and contracted in unison; however in practice, due to the difference in metal materials and thicknesses, there is a lag in the effect of temperature changes and a consequent potential loss in accuracy.

The system is designed to maintain uniform temperature throughout day and night operations. It is expected that the maintained tem-

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GAGES—As soon as the national defense program began to take shape, Greenfield Tap & Die Corp. set about increasing its productive facilities, and through the coöperation of the government, a new plant is in operation today—a plant which will more than quadruple the company's production of vital tools and at the same time assure a closer accuracy control despite the greatly increased quantity and speed of production. . . . The basic requirements of the building dictated the type as well as design of the important air conditioning system, viz., mechanical ventilation, elimination of atmospheric and manufacturing dust, and stabilization of temperature. This latter is most important, as all hour to hour fluctuation of temperature which causes expansion and contraction of metal has to be eliminated. This stabilization is maintained day and night and even from season to season the change will be extremely slight. . . . One air conditioning system, consisting of fans, filters, heating and cooling coil assemblies with pumps, ducts and accessory equipment, controls the temperature and humidity in the building proper. The control system is completely automatic. On the second floor, there is a final inspection room, 60 ft long by 40 ft wide, separately air conditioned. It is maintained at 68 F temperature and 50 per cent relative humidity in accordance with the recommendations of the Bureau of Standards. . . . Mr. MacNeish, of McClintock & Craig, Inc., the engineers and architects, describes the air conditioning here

perature will be adjusted weekly or monthly during week ends or other shutdowns changing by increments with the seasons from 68 F in winter to 78-80 F in summer. In this way the most economical operation can be obtained with no loss in the accuracy of manufacture.

Air Distribution

For the building system there is one central fan, filter, heating and cooling coil assembly on each of the two floors. To the first floor, 20,500 cfm of air is supplied at a maximum differential in temperature between entering air and room air of 20 F; 16,873 cfm is recirculated air and the balance of 3627 cfm is outside air. This provides 36 cfm of outside air per occupant and makes six complete changes of the room air per hour.

Similarly the second floor is supplied with 14,700 cfm of air, the temperature differential between entering air and room air is 15 F, and the outside air supply is 3627 cfm or 36 cfm per occupant. There are four changes per hour of the air in the second floor.

The air supply system for both floors is similar. The return air from the room and the outside air combine and pass through an automatic self-cleaning air filter, then through evaporator cooling coils into the suction of a motor driven centrifugal fan. The discharge air from the fan splits into four duct systems serving the four zones into which each floor is divided. To control the temperature in the four zones, reheat steam coils are installed in the discharge ducts.

Air is supplied to the room through air registers arranged for manual adjustment of the volume and velocity of the air stream. Each of these registers serves approximately 400 sq ft; they are situated on the duct sides on 20 ft centers.

There is a continuous duct along the long north wall of each floor for the return air. It is substantially constructed of stretcher levelled steel plate. This continuous duct on the wall of most severe winter exposure avoids a zone of cold air along this wall, permitting production work to be performed close to the wall and allowing maximum utilization of the available floor space.

Direct Expansion Refrigeration

The refrigeration system uses "Freon 12" and operates on direct expansion. Two compressors in the basement discharge to evaporative shower type condensers on the roof. From the condensers, the liquid refrigerant passes down through expansion valves into the cooling coils in the air supply duct for each floor. From the coils the refrigerant gas returns to the compressors, completing the cycle.

The control system is completely automatic. There are four thermostats on each floor, one for each of the zones. The thermostats regulate the steam supply to the reheat coils to maintain constant temperature in the zones. The thermostats, during the cooling season, open or close in rotation the expansion valves on the four cooling coils of each floor.

The two compressors, each four cylinder 6 in. x 5 in. machines running at 485 rpm operate on suction pressure, each compressor having three stages of capacity: 50, 75 and 100 per cent. The calculated total load of the building system is 83.5 tons.

Final Inspection Room

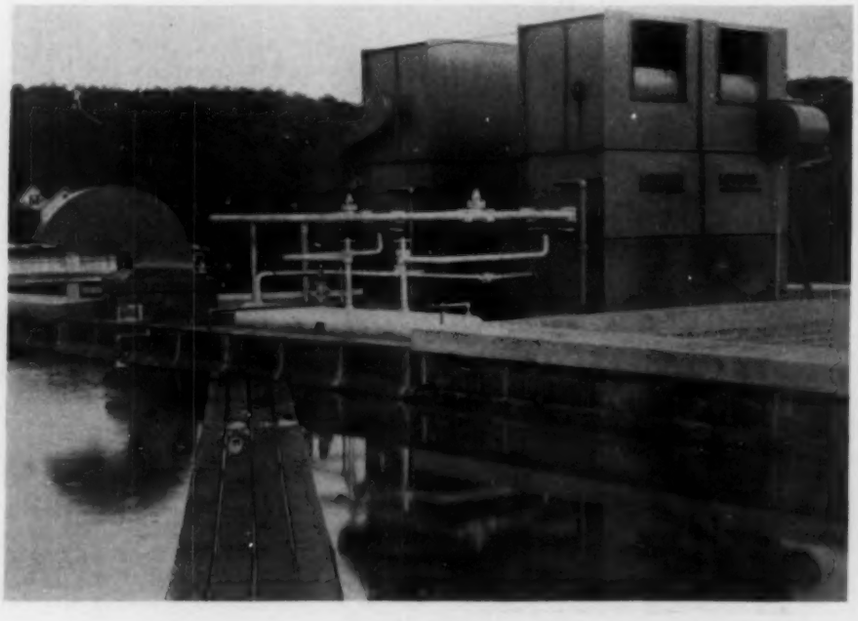
On the second floor of the gage building, there is a room built with tile walls to which there are two entrances; each entrance has a vestibule and an inner door. The room is 60 ft long by 40 ft wide and is centrally situated so that none of its walls are adjacent to the main building walls. This is the final inspection room, and it is separately air conditioned.

In this room the dry bulb temperature is maintained at 68 F with variations not exceeding ½ F; relative humidity is maintained at 50 per cent with variations not exceeding 3 per cent, equivalent to ½ F variation in wet bulb temperature.

In this room the final inspection of the finished product is made. The gages are brought into the room and allowed to stand for some time to allow the metal to reach the room temperature, after which the accurate measurements required in this operation are made.

The air condition of 68 F temperature and 50 per cent relative humidity is in accord with the recommendations of the Bureau of Standards, as well as being the recognized international standard at which all such work should be calibrated.

Five thousand cubic feet of air per minute is delivered to this room through six ceiling type air diffusers, the maximum temperature differential between entering air and room air being 6¼ F.



The two compressors for the *building* system discharge to evaporative shower condensers on the roof; from the condensers, liquid refrigerant goes to the cooling coils in the air supply ducts for each floor. Water from the water cooled condenser for the *final inspection room* air conditioning system is discharged to these evaporative condensers on the roof. Surplus water is used for flooding the roof to reduce heat gain through it

A duct built of stretcher levelled steel plates around the walls exhausts the air uniformly from all the room. There is a complete air change in this room every six minutes.

A return air duct of heavy metal construction is installed along the north wall of each floor in order to avoid drafts due to the exposure in cold weather

"Outside" air can be alternately taken from outdoors or from the building conditioning supply ducts. Outside air and return air from the room combine in a plenum chamber and pass through an electrostatic air filter, then through an air washer and a heating coil into a motor driven centrifugal fan, from where it is discharged through ducts and diffusers to the room.

Chilled water is pumped through the air washer. To control dew point and humidity, automatic control maintains this water at a constant temperature until there is a change in the relative humidity in the room and then a humidistat in the return air duct resets this control to a higher or lower temperature as required. Similarly, a control maintains constant dry bulb temperature in the air discharge by admitting steam to the reheat coil. A room thermostat resets the constant temperature control. Dry bulb temperature and relative humidity are continuously recorded on charts inside the room.

For cooling, water from the air washer is pumped through a water cooler in the basement and back to the air washer sprays. Part of the water is bypassed around the cooler and mixed with the chilled water in an automatically controlled three-way valve to maintain the desired temperature.

The water cooler is served by a motor driven four cylinder 4¼ in. x 3 in. compressor operating at 550 rpm, with a water cooled condenser.

Water from the condenser is discharged to the evaporative shower type condensers on the roof which serve the two compressors on the building system. Surplus water is discharged to the main roof for flooding. The maximum calculated heat load of the final inspection room is 14.4 tons.

The air conditioning systems were designed by McClintock & Craig, Inc., engineers and architects, who also designed the entire building and other services. E. B. Howell Co. was the successful bidder and made both installations.

UPPING EFFICIENCY WITH AIR CONDITIONING

Blemishes in newspicture negatives due to variations in developing solution temperatures have been practically eliminated in the modern photo laboratories of Philadelphia's Evening Bulletin, since air conditioning was installed. In addition, elapsed time from exposed negative to final print has been cut down to 7½ min with the help of air conditioning and other modern devices.

The problem of quick photographic processing is an acute one in these days of fast picture work and high speed news publishing. Rapid developing, fixing and washing of prints has been achieved in the completely air conditioned dark room through the introduction of up-to-date methods. Even hot air and fumes generated by carbon arcs used in the printing room are removed through forced ventilation. Films are dried in a centrifugal dryer. Electrically heated ovens are used to dry prints not marked "rush."

Air conditioning was first installed at the Bulletin in 1933, when the city room and editorial offices were modernized. In 1936, the paper's ground floor public space was air conditioned. A year later, a modernization program involving the entire building was launched.

Cooling equipment includes eight refrigeration units with a total cooling capacity of 350 tons. One of the unusual features of the installation was that the refrigeration machines had to be situated in a circular clock tower atop the 10 story Bulletin building. An unique "spider web" foundation of structural steel was built to provide suitable vibration and sound isolation. Charles S. Leopold is the consulting engineer and George Howe is the architect.—EDWARD T. MURPHY, vice-president in charge of marketing, Carrier Corp.

G. E. CONDITIONS ITS NEW BLACKOUT PLANT

IN the "blackout" type supercharger factory the General Electric Co. has completed at Everett, Mass., an air conditioning system will be operating for cooling even after the snow of a New England winter lies heavy on the ground. The factory is another of the vital defense plants which the company has built as agent for, and in behalf of, the Defense Plant Corp., subsidiary of the Reconstruction Finance Corp.

The major portion of the internal sensible heat load in this new 400 by 500 ft windowless factory consists of heating equipment and power machinery used in the manufacture of the airplane turbo-superchargers. As the building is well insulated, the amount of heat released by this equipment when the plant is operating at full capacity, plus full fluorescent lighting, is sufficient to heat the building when the outdoor temperature is only 15 deg above zero. Consequently the air distribution system has been sized and located to balance the internal load on cooling, with only small modifications at points of maximum heat release to allow for some lack of balance between heating and cooling loads.

The manufacture of turbo-superchargers involves numerous very accurate machining processes on

ALUMINUM AND MAGNESIUM ALLOY MACHINING PROCESSES REQUIRE ACCURATE TEMPERATURE CONTROL

aluminum and magnesium alloys, which require close temperature control throughout the year for successful results. This had to be the primary factor in the design of the air conditioning system. As far as possible, however, every effort has been made to maintain, at the same time, air conditions within the comfort zone.

The total capacity required by the system is 1140 tons of refrigeration, which is provided by eighteen 60 hp, eight cylinder reciprocating condensing units and direct expansion coils. The 18 units facilitate flexible control by matching load variations at various times and in different parts of the plant. There are six fans which serve six independently controlled zones in the plant. All major equipment is situated in three mezzanines centrally located with respect to load and built into the building structure, so that all machinery, ductwork and outlets are above the 18 ft level. Each mezzanine has two machine rooms containing the three compressors, fans and coils for the zone served from this space.

The system is designed to produce a maximum of 11 deg of cooling at an outdoor temperature of 91. Controls are of the pneumatic type. Outside air is introduced at each mezzanine through the roof directly above at the rate of 100,000 cfm, except during cooling or heating load peaks. At such times it is reduced to one change per hour. As the building has no windows and only a very few openings, full exhaust capacity for the entire amount of outside air introduced is provided by 18 exhaust fans situated under the eaves of the building. All air in the building is cleaned by automatic air filters operating continuously; with full time operation, there would be no opportunity to change dirty filters.

Air is distributed to the six zones through ductwork of standard construction and ceiling discharges of the venturi bowl type situated at the bottom of the building trusses about 18 ft from the floor.

Heating is accomplished with steam coils modulated in six steps for each zone and requires a maximum outlet temperature of only 25 deg above room temperature under design heating conditions. The air outlets are situated in the center of each 40 ft bay with a downward angle discharge sufficient to provide thorough mixing of the discharge air at the floor level both under heating conditions and under cooling conditions with a discharge temperature 20 deg below room temperature.

A 1200 ton cooling tower is situated approximately 90 ft away from the building. Cooling water is circulated to the condensers by a steam turbine driven pump, the exhaust steam from which is used in the manufacturing processes. All piping is cast iron outside of the building and welded steel inside. Piping is distributed to the condensers through tunnels under the floor of the manufacturing space, with risers running up to the three mezzanines along the columns of the building.

An interior view of part of the $5,000,000 factory nearing completion. Air conditioning ducts may be seen in the ceiling. Walls of factory are tiled; lighting is fluorescent

PROPERTIES OF PLASTIC PIPING

William H. Adams, Jr., Reviews Design and Installation of Molded Phenolic Resin Piping

PLASTICS—much in the news these days—have their applications in the industrial piping field. In this brief review, Mr. Adams, technical director of the Haveg Corp., explains the properties of a type of pipe comprising a molded phenolic resin asbestos composition. He gives design and installation data, tells where it should and shouldn't be used

THE problem of pipe lines for the handling of corrosive liquids, gases and fumes is a source of difficulty in many industries. Engineers are frequently uncertain as to the best and most economical material to use for a given application, and even when a material has been selected, the conditions which govern its design, installation and maintenance may not be well understood.

The purpose of this article is to answer such questions as these for one type of corrosion resistant material, a phenolic resin plastic. Such material has been used for many years in industrial equipment to combat chemical action, and as a result, fairly accurate predictions can usually be made as to its applicability for a given service.

The type under consideration is a molded phenolic resin asbestos composition. The resin is mixed with an acid washed asbestos which has excellent acid resistance. It is then molded by a unique process. Because of the use of special asbestos as a filler, the chemical resistance of this composition is uniform throughout the thickness. Consequently, it is possible to apply it in very severe corrosive service, even under hot conditions. It can be used at temperatures as high as 265 F and is not subject to thermal shock. Boiling acids like hydrochloric for instance, can therefore be handled.

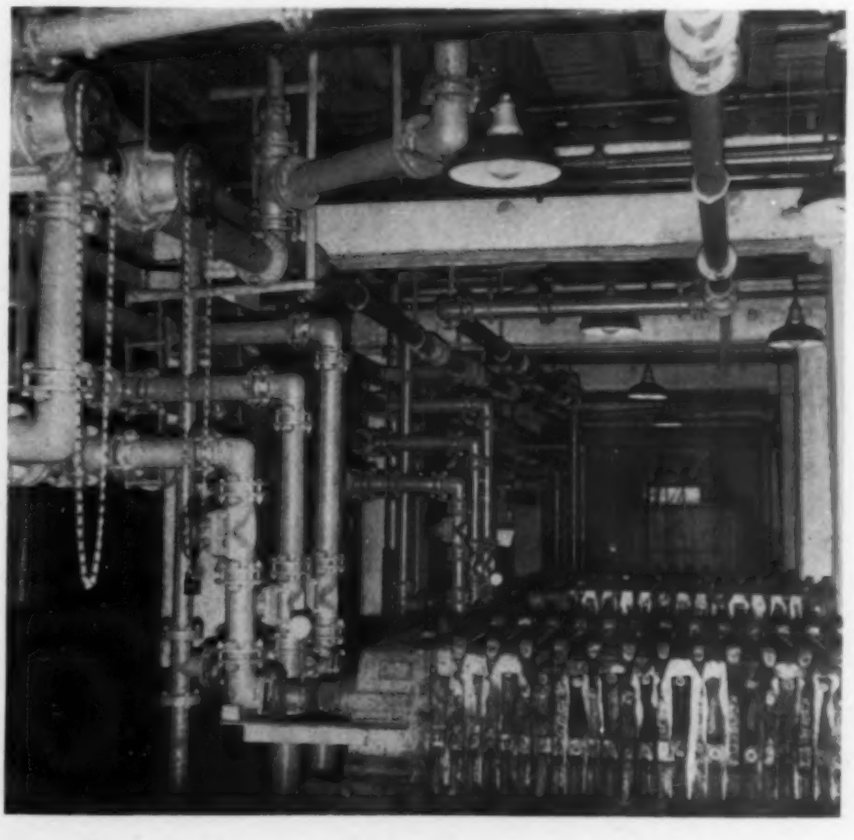
Chemical Resistance

This material is resistant to practically all acids except oxidizing acids like nitric or concentrated sulphuric (higher than 50 per cent)—and by "resistant" is meant even under hot conditions (up to 265 F). It is resistant to salts, most of the weak bases like sodium carbonate and ammonia, to chlorine, and to many solvents, especially hydrocarbons. It is not resistant to strong bases like sodium hydroxide, nor can it be used with oxidizing agents, acetone, pyridine and other similar solvents. It is satisfactory for use with sulphuric acid at concentrations usually used in pickling steel.

Friction Loss

The size of the piping required will, of course, depend upon the factors of allowable pressure drop, quantity of fluid, temperature, viscosity, friction factor, etc. All of these except the friction factor are well known and the expressions for their calculation are readily available. The friction factor is, however, needed for an accurate design. Unfortunately, data on this point are very meager to date. Tests which have been made indicate, however, that friction losses are of the same order of magnitude as for smooth iron pipe. The best procedure for the present seems therefore to be to calculate the size of iron pipe required, and then possibly employ a small safety factor.

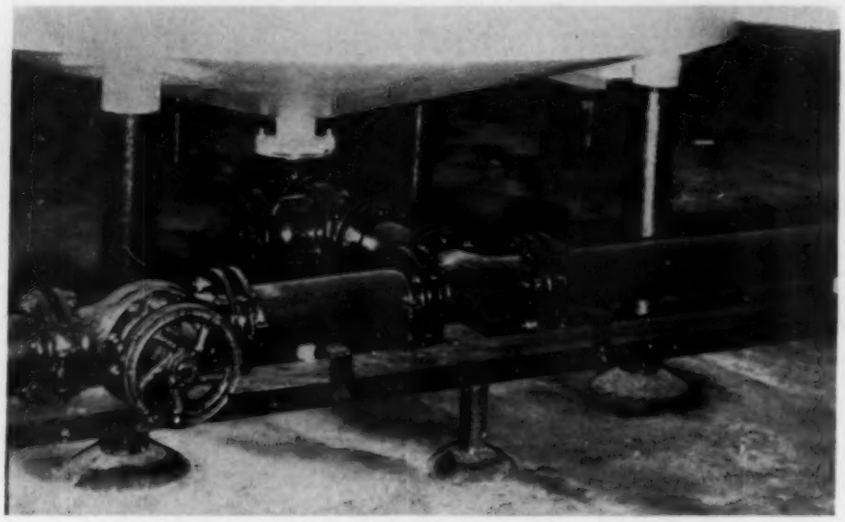
The sizes regularly produced are from ½ in. to 12 in., with larger sizes possible as specials. The lengths are ordinarily 10 ft. Con-



Part of a large installation of plastic piping in a chemical plant

nections are made by means of split cast iron flanges which set in slots machined near the ends of the pipes. Bolt circles are the same as U. S. Standards, so that direct connections with other types of flanged pipe are possible. Diaphragm type valves are employed, made with a body of the same composition as the pipe.

For handling gases or fumes, light wall pipe in large sizes is available. This is regularly made up to 30 in. diameter or larger for special orders. It is connected either by flanges similar to those used on the regular pipe, or by means of bell and spigot ends.



Close-up of plastic pipe and valves controlling flow from large storage tanks

Pressure Requirements

Such pipe is seldom called on for service under heavy pressures because these are infrequently required with highly corrosive chemicals. The normal pressure maximum is about 60 psi, depending somewhat upon the temperature and nature of the solution. At the extremes of temperature (from 212 to 265 F) the pressure should not exceed 25 psi, at least on the larger sizes.

The light wall fume duct of this material is not designed to resist either much pressure or vacuum. A few inches of water is all that is recommended. If gases or vapors at appreciable pressures must be handled, pipe should be used.

Installation and Support

Installation presents no particularly difficult problems, but a full understanding of the nature of the material is necessary for best results. In brief, this material is a plastic, and must be handled as such. This means careful physical handling, elimination of spring fits, use of even and light pressures in tightening joints, and the provision of adequate and well lined up supports.

The piping is supplied either unmachined (in lengths of 10 ft) or cut to size and with the ends machined for the flanges. Most customers order at least some of the lengths unmachined in order to permit accurate fits in the field. The pipe can be cut with an ordinary hacksaw and can be machined in any lathe which has sufficient length. Fittings are nearly always supplied with the ends already machined.

The actual connection is made by bolting together the cast iron flanges, using a gasket between the pipe ends.

Supports for horizontal lines should consist of straps at each side of the joint, plus one between joints. Longitudinal movement should not be inhibited. In vertical lines the individual pipes should be separately supported by straps or in a similar manner, in order to avoid resting the entire weight on the lowest piece.

Expansion and Contraction

Phenolic resin compositions have a rather high rate of expansion, which means that provision for expansion joints should always be considered. The coefficient of expansion will be ordinarily about:

$k = 32.7 \times 10^{-6}$ per degree Centigrade.
$k = 18.2 \times 10^{-6}$ per degree Fahrenheit.

The actual elongation of a given line can then be calculated by:

$$e = k \times l \times (t_1 - t_0)$$

where k = coefficient of expansion;
l = length, in.;
e = elongation, in.;
$t_1 - t_0$ = final and original temperature differential.

If both ends of the pipe line are quite free to expand and contract, we would expect not to require any expansion joints. Experience seems to indicate, however, that an expansion joint every 100 ft or so is desirable in long lines to prevent expansion difficulties. If the ends of the line are fixed, however, full provision for expansion must be made.

Two general types of expansion joints are regularly employed. The first is a simple bell and spigot joint, packed preferably with an elastic material. The second consists of a sleeve and gasket assembly which permits considerable expansion. The two pipe ends are installed in the sleeve a little apart, thus permitting longitudinal movement in either direction. A single such joint can take up as much as 1½ to 2 in. expansion.

It is frequently rather time consuming to calculate the exact expansion requirements of every branch from the above formula. The following general rules will in fact suffice for most cases:

1) If both ends of pipe are fixed, at least one expansion joint is required, no matter what the length of pipe.

2) In a pipe line with fixed ends, the use of two expansion joints per 100 ft will take up the expansion in all but extreme cases.

3) In lines in which one or both ends are free, an expansion joint every 100 ft will prevent difficulty from expansion.

Maintenance

The maintenance of a properly installed pipe line in the right service is not a difficult matter.

First of all, joints should be kept tight; but, if joints are tightened, provision should be made for the change in length which will result. An expansion joint in the line will allow for this.

Supports should be checked from time to time and adjusted if the alignment or bearing is no longer correct. Valve diaphragms should be examined periodically and replaced if they are seriously worn.

Conservation for National Defense

THERE IS NO SUBSTITUTE FOR FUEL

. . . . . therefore, its efficient use is essential. Kalman Steiner discusses methods for heating plant operation

NEWSPAPER cartoonists usually find—on one pretext or another—at least one occasion during the average winter for running a picture of Mr. John Q. Public toiling to keep ahead of the voracious fuel appetite of that monster in his basement called the furnace. This perennial topic of the newspaper illustrator is once more front page news, and bids fair to crowd the headlines for many months. Before next winter is over we may all be more concerned about fuel supply than about priorities on tinplate. Human ingenuity can think up substitutes for lots of things that otherwise were considered indispensable, but no one has yet come forward with a substitute for fuel. We have only to think back to the fuelless days of the last war to realize how important a commodity fuel can be.

NO SUBSTITUTE—Before next winter is over, we may all be more concerned about the fuel supply than about priorities on tinplate. Thinking back to the fuelless days of World War I, and remembering that no one has come forward with a substitute for fuel, it can be appreciated that the best and most economical use of this indispensable commodity is highly important. As a group, plant and building operating engineers can do more than any other group to conserve this vital national resource . . . Mr. Steiner—of the Ace Engineering Co.—discusses here methods of operating heating systems to utilize the fuel most effectively. A heating system is inherently wasteful, for it must be sized for the maximum load. Therefore, good operating practices, the right control equipment and reductions in the building heat requirements always will pay high returns

I pointed out in a recent article in HPAC that among the chief needs of the nation during the defense emergency is a vigorous and effective movement to promote fuel economies. Operating engineers—more than any other single group—can do much to assist in such a program, because great quantities of fuel are burned in the plants under their charge. Possible fuel economies fall into two classifications: steam generation and steam utilization. In the August issue there was presented a resumé of the steps to be taken, particularly in oil fired boiler rooms, to assure minimum fuel consumption in making steam, and this month an attempt is made to submit some thoughts on reduction of steam used in heating. (It should be understood that as used here the word "steam" is a generic term for "heating medium," and implies hot water and often warm air as well.)

Need a Speedometer

If we consider a heating plant as a mechanical contrivance, it might be said of it that the rate at which it goes probably receives less thought than the rate of propulsion of almost any other kind of contrivance. There are all sorts of speed indicators and speed governors, applicable to all sorts of mechanical systems. Yet no very simple or universally applicable speedometer has been devised for a heating system, and as a result the speed at which one is driven is too often purely the result of chance or whim. We all appreciate, of course, that when it goes too slowly irate tenants promptly complain to building managers and engineers, but rarely does one hear of a tenant complaining of receiving too much heat.

Heating Plant Inherently Wasteful

So it happens that the system which has most need for a reliable rate index is often denied one. There are few systems that can be as wasteful as a heating plant, and that

fault is inherent to all heating systems. The extreme variation in outdoor temperature brings this about. We must design for coldest weather. To again draw a parallel with other mechanisms, load carrying devices are also designed for maximum load, but need not frequently be operated at much less than maximum. Thus, a 10 ton truck is seldom called into service to deliver a spool of wire or a length of 1 in. pipe; other transportation is provided or deliveries are allowed to accumulate until the large truck can logically be used. This is not possible with a heating plant. Heat must be provided from the same plant whether the load be 5 degree days or 75—whether the outside temperature is 60 or —10. And business men who would shudder at the folly of the trucking exaggeration mentioned are often oblivious to the fact that their heating plants are operated in exactly that fashion.

But if carelessly operated heating plants are wasteful, it is equally true that every economy accomplished pays a high return. Since combustion efficiency will range between 60 and 75 per cent in the average case, every Btu saved in diminished steam consumption represents from 1.67 to 1.33 Btu in actual fuel saving.

Importance of Control

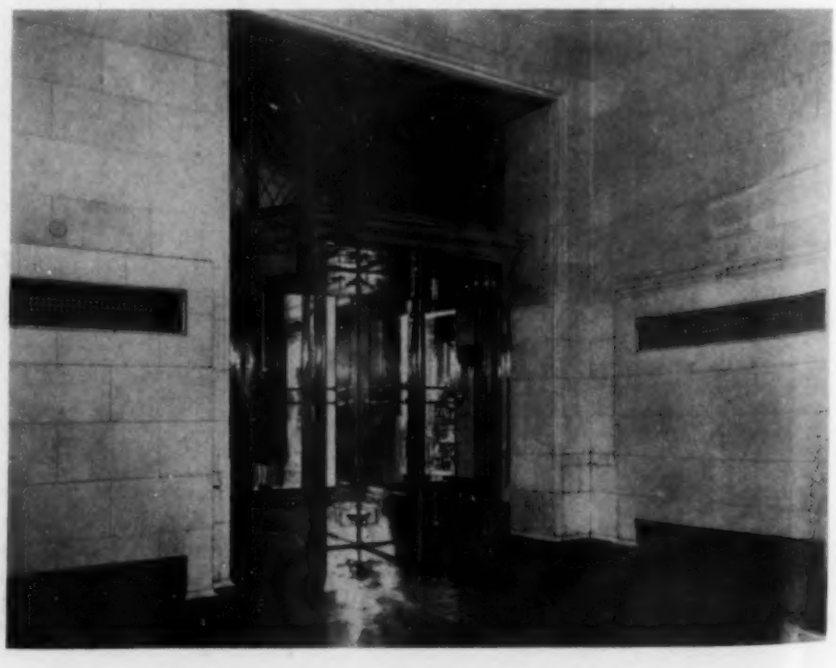
Many years ago manufacturers of oil burners, stokers and gas burners learned of the excellent sales possibilities of their devices when they were offered to the public as automatic heating rather than as assemblies of nuts, bolts, and gears. Working in conjunction with engineers of the control manufacturers, some really fine products were developed for providing automatic home heating with the various types of heating plants. With this splendid record of success in residential heating, it is remarkable to find that when these same fuel burners were applied to the heating of larger buildings a strange apathy sometimes set in, and because the common room thermostat did not seem to lend itself to use in a large building, all thought of temperature control was abandoned. Thus we find many installations where the burner or stoker will simply keep a constant steam pressure on a boiler, regardless of the outside temperature.

Attempts soon were made, however, to place a measure of control over heating equipment. Typical of free-lance design were such devices as aquastats placed on condensate return lines for the purpose of stopping the burner when the temperature of the return water indicated that steam had reached most portions of the heating system; an aquastat immersed in the boiler water to lock out the burner until the water temperature had fallen to a certain minimum after the burner had been stopped by the pressurestat; a thermostat set outdoors to interrupt burner operation (which otherwise was under pressure control) whenever the weatherman delivered 65 degrees or better; a thermostat in the janitor's quarters of an apartment or the front office of a hotel, enabling the janitor in the first case and the clerk in the second to temporarily raise the heat level if a chilly tenant asked it.

But there were also more fundamental efforts being devoted to the problem, and gradually there evolved a number of really fine controls and control methods. Today no one need grope around or improvise; a completely developed device or system is available for proper automatic regulation of any type of heating plant. These range in complexity from single switches cutting the firing device in and out in accordance with building steam requirements to elaborate systems involving zoning of large buildings into smaller sections, which in turn are each controlled as individual units by means of steam valves, temperature controls, timing devices, pressure and flow regulators, indicating and recording instruments.

Defects May Show Up

Once a heating system is placed under automatic control, some defects that otherwise might not have been noticed may manifest themselves. Steam or condensate piping improperly pitched may give passable results when the steam flow is unrestricted, but under controlled flow pockets of water may collect and retard free circulation. Such faulty piping will have to be corrected, of course. One of the commonest locations of such pockets is in the radiator runout connection from the riser. Another common defect is improper venting of gravity heating systems, both on the mains and on the radiators. Automatic control can only be successful



"We are all familiar with the usual precautions taken against unnecessary heat losses: weatherstripping of windows, caulking of window frames, closing off shipping spaces from the rest of the building, use of revolving doors or double vestibule doors, letting unoccupied space remain unheated, covering steam pipes in basement and attic spaces"

if steam circulation is uniform and rapid. On two pipe systems, it may be found desirable to orifice the radiators.

Reducing the Heat Needed

So far the discussion has covered the normal or regular heat requirements of a building. In many cases that heat requirement can be reduced. We are all familiar with the usual precautions taken against unnecessary heat losses: weather-stripping of windows, caulking of window frames, closing off shipping spaces from the rest of the building, use of revolving doors or double vestibule doors, letting unoccupied space remain unheated, covering steam pipes in basements and attic spaces, etc.

The ventilation laws of most communities set up rather generous air change requirements, intended to meet the most severe tax on them that might arise. Generally, operation of such systems at peak outputs circulates a great deal more air and provides ventilation far in excess of reasonable requirements. Great care should be exercised in allowing these systems to run without regulation, as large amounts of heat may be needlessly rejected from the building in this way.

There are often cases where a simple change in piping may effect considerable heat savings. First floor stores in multi-story buildings may be the only portions of a building requiring heat after regular hours. These stores should be piped independently, so that the remaining space need not be heated during that time.

Perhaps the best closing thought might be to direct attention to the possibilities of insulating walls and roofs. If the building construction lends itself to such improvement, it will generally be found that the cost is quickly recovered.

Many engineers have known that certain improvements to their heating plants and buildings would result in appreciable fuel savings, but have been unable to secure funds for making the improvements. Today it becomes a matter of public duty. Our fuel resources must be conserved, and there is no better place to start than by eliminating waste in heating. If voluntary action now is sufficiently effective, it may obviate rationing at some later date.

HOW DOES SOLAR HEAT GAIN THROUGH FROSTED AND PLAIN GLASS COMPARE?

THE EDITOR—

I would appreciate any available data on the solar heat gain through "frosted" glass as compared with plain glass.—V.B.

REPLY (by Dr. F. C. HOUGHTEN, director, ASHVE research laboratory)—In reference to solar heat gain through frosted glass as compared with plain glass, "V.B." is no doubt acquainted with the paper on Heat Gain Through Western Windows With and Without Shading which appeared in the ASHVE Journal Section in the April, 1941, HPAC. Fig. 3, particularly, of that paper gives the heat gain through plain windows with southern and western exposures. These are without any sun arresting appurtenances whatever. The subsequent figures in this paper give the effect of various sun arresting appurtenances on western exposed windows together with a little information on solar heat gain through heat absorbing glass.

Answering the question directly, we have never included frosted glass in our studies, nor are we acquainted with any other studies of this phase of the subject. However, from the accepted theory of performance of frosted glass, we should be able accurately to predict the behavior of solar radiation through it. Frosting as generally accepted involves only the roughening of one or both of the glass surfaces. This roughening may be had by rubbing with a fine emery cloth, etching with acid, sand blasting, or occasionally, I understand, by placing a film of some other material on the surface. In the latter case much may depend on the character of the film or material placed on the glass. However, since it is very thin it probably does not absorb much of the solar transmission.

With the other forms of frosted glass where the surface of the glass itself is only roughened, the effect is to place a great many prismatic surfaces, heterogeneously oriented, so that the light rays are bent in devious directions at the surface; the result is that the light does not come through that surface in parallel rays, and hence one cannot see an object through it.

However, it is to be assumed that all of the radiation and therefore all of the heat does get through. In fact, this may be assumed from another line of reasoning; namely, the fact that frosted glass is assumed not to reduce the visible light passing through the glass or the illumination resulting.

So on the whole, I believe that it is safe to assume the heat gain through frosted glass by solar radiation (or by transmission either, for that matter) will not differ materially from that for plain glass.

THERE OUGHT TO BE A LAW

Compulsory Cooling?—Contending that more people die of heat prostration than actually die of exposure and cold, Dr. Willis H. Carrier predicted last month that "compulsory cooling" legislation will be enacted in America in the not too distant future. Legislation will come first in the "hot industries," he predicted, particularly in those cases where extremely excessive working conditions tend to impair the health of the laborers.

UNIT HEATERS DIRECT RADIATION ZONE CONTROLLED

. . . . . one pipe, zoned, forced circulation hot water system heats new Bell & Gossett plant

AS noted on the Editor's Pages of our August number, the Bell & Gossett Co. is erecting a new, greatly enlarged plant at Morton Grove, Ill., a suburb of Chicago. The factory is on a 20 acre piece of ground, which allows room for future expansion. In designing the building, the best practices of straight line production were taken into account. Raw materials enter at one point, emerging with a minimum of handling as finished products.

The factory itself contains 70,000 sq ft of floor space. Its mill construction roof is supported on concrete pillars and is specially treated to give a maximum of insulation against heat and cold. A novel wall construction is employed; hollow bricks, laid double, provide insulation because of their air spaces. Shipping and receiving platforms will accommodate five truck trailers at a time and are completely enclosed to prevent needless waste of heat during the winter. An addition to the main factory building houses the paint spraying and cleaning departments, thereby eliminating the spread of noxious fumes.

Forced circulation hot water supplies heat to the building, and the system is zone controlled to allow greatest flexibility of heat distribution and to permit full compensation for occupancy factors. The office can thus be kept at the proper temperature for sedentary workers, while other portions of the building are held at temperatures best suited to the nature of the work. The executive offices are on one zone; the main office on another zone; the engineering department, factory locker room and the shower room are on the third zone; the advertising, shop control and service departments are on a fourth zone, and the factory is on a fifth zone. With this arrangement, if only one or two departments are working overtime, those zones can be maintained at their regular temperature, while the others drop to a lower temperature.

Various types of unit heaters and direct radiation are used. Direct radiation is specified throughout the office with the exception of the engineering department and the factory locker and wash room, where panel heating with pipe coils in the concrete floor is provided. Both horizontal and vertical unit heaters serve the factory proper.

Lower temperature water will be circulated through the pipe coils in the floors than through the radiation zones. It will be controlled automatically by coupling the pipe coils to the heating circuit with a converter and controlling the temperature of the water automatically to match the outdoor temperature. A similar operating control arrangement will be used for the heating of the main office radiation circuit, while the other zones will be operated by zone thermostats connected to their respective zones.

The boiler water will be evenly distributed throughout the radiation circuit by use of a one pipe system using diverting tees. All of the radiation on the first floor is served by an overhead main, using bottom outlet fittings for connecting each of the radiator units.

The two automatically fired steel boilers are equipped with submerged heaters for heating service water instantaneously, without the use of a storage tank.

Architecture and engineering is being handled by the firm of Danely and Weidemeyer.

HOT WATER SYSTEM — A one pipe, forced circulation, zoned hot water heating system will serve the new Bell & Gossett plant at Morton Grove, Ill., to be devoted to the manufacture of hot water heating systems and specialties. Various types of direct radiation are used in office areas, both vertical and horizontal unit heaters serve the factory proper, and in certain areas prefabricated pipe coils in the concrete floor will provide panel heating. . . . Water temperatures will be automatically controlled in accordance with variations of outdoor temperature

Condensation in Compressed Air Piping

BY G. M. BROWN, MECHANICAL AND ELECTRICAL ENGINEER
FOR CHARLES FOSTER, CONSULTING ENGINEER

CONDENSATION of moisture inside compressed air piping systems is often rather mystifying. It is believed that the chart reproduced on the next page may give a means of visualizing the relation between the formation of water vapor and the prevailing pressure and temperature conditions in compressed air systems.

Since this chart is very similar in nature to psychrometric charts, there seems to be little need for lengthy explanation. However, in order to illustrate a surprising fact and one which will be of particular interest to those concerned with compressed air services, the following example is given:

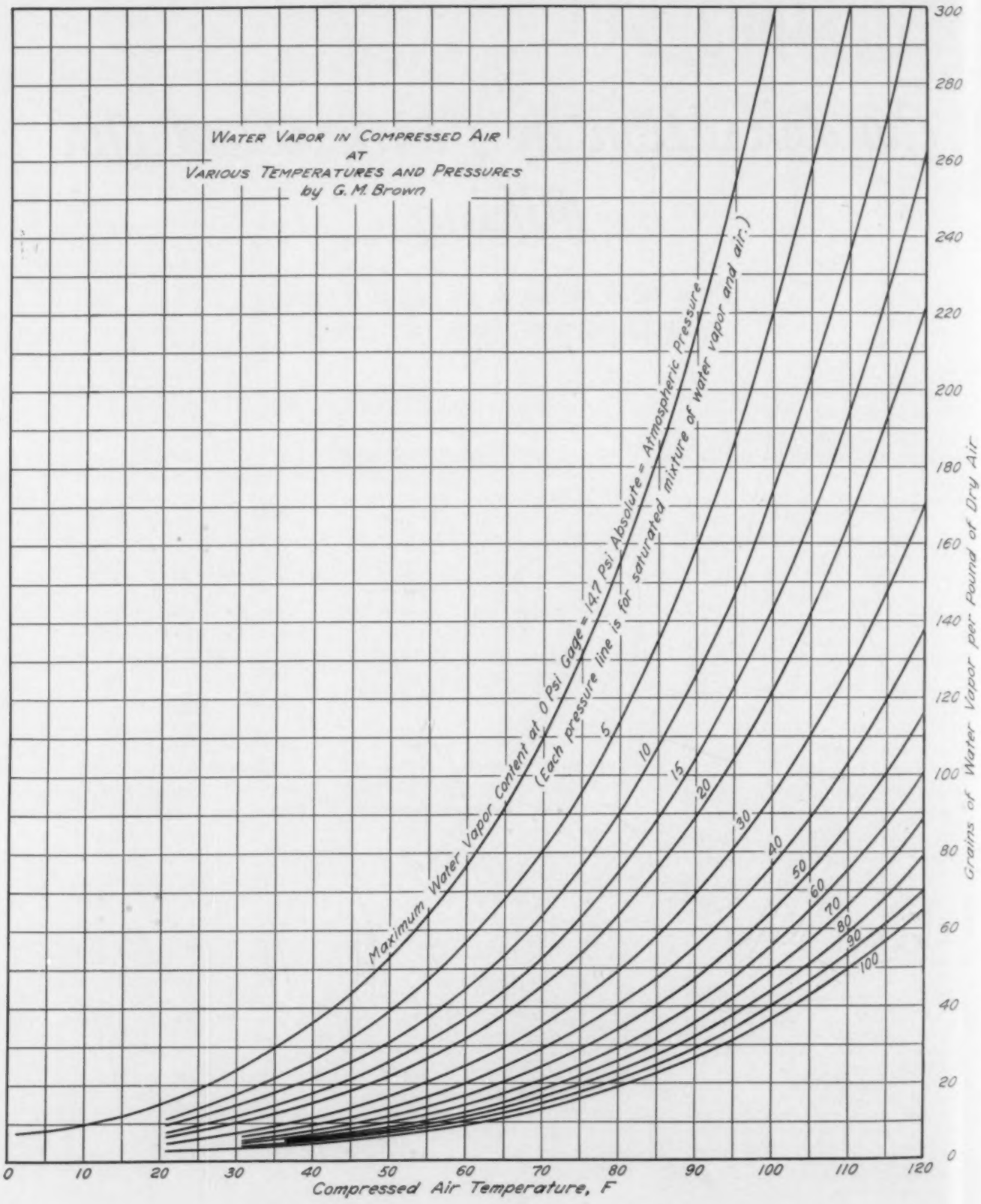
Suppose 50 F air having 40 grains of water vapor per pound of dry air (this corresponds to approximately 76 per cent relative humidity) is compressed to 30 psi gage pressure and is delivered to a tank which is maintained at 70 F. Some of the water vapor will condense out because, when saturated, the air under 30 psi gage pressure and 70 F temperature, can contain only 37.5 grains per pound of dry air. Now if this air is allowed to expand to 15 psi pressure and is passed out through pipe lines which cause it to cool to any extent, condensation will again begin to form when the air at this lower pressure reaches a temperature of 58 F.

Now these above conditions are by no means improbable in a compressed air piping system and far more unfavorable conditions could have been selected for illustration.

A study of the chart will illustrate certain desirable features of a compressed air piping system: First, air should be compressed to a pressure considerably in excess of the reduced pressure at which the compressed air is to be used; second, the high pressure tank should be maintained at temperatures as low as possible; third, the tank should be automatically blown off to prevent accumulation of moisture; fourth, no compressed air piping should be run through spaces where the air may be cooled to the saturation point for the particular pressure. Tendencies to comply with all these requirements are demonstrated in many installations, but their importance is not always fully realized.

HEATING, PIPING AND AIR CONDITIONING DATA SHEET

See preceding page for explanation of use



Heating·Piping·Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

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Use of Cold Accumulators in the Air Conditioning Field

By R. W. Evans* and C. J. Otterholm**, Minneapolis, Minn.

Section I—Introduction

THE cold accumulator was first developed to store *refrigeration effect* to meet the short period refrigeration demands, found in the Dairy Industry. Similar needs for a refrigeration *storage battery* are found in many types of air conditioning work.

Considerable data are now available for the determination of peak cooling demand. These, however, are insufficient for the selection of accumulator equipment, since such selection is dependent upon the total cooling demand in a 24-hour period.

This paper concerns the introduction of new factors, such as, *Accumulated Degree-Hours* and *Accumulated Sun Intensity Values* to be applied in the accurate determination of total cooling load. The performance characteristics of one type of cold accumulator and its application to air conditioning loads of several types is also discussed, together with certain specific economies obtainable with this method of cooling.

Section II—Performance Characteristics of a Typical Cold Accumulator

The type of accumulator discussed in this paper originated as a plan to utilize water as a substitute for brine in dairy and creamery plants. A very satisfactory unit was developed and based on the principle of using thin sheets of ice, formed on copper plates, with high rates of water circulation. It was found possible to cool milk and cream to 40 F just as quickly with 35 F water plus high velocity circulation as was ordinarily obtained with 20 F brine. Over 200 of these cold accumulators are now in daily operation in the United States and Canada.

The air conditioning accumulator presented no new problems in design, except those relating to increased storage capacity. Due to the wide range of air conditioning loads, requiring corresponding wide ranges of storage capacity, it was decided to standardize on one size unit, having a nominal storage capacity of 24 ton-hours of cooling effect, and installing as many units or sections, as may be required to meet any air conditioning load requirement. The manufacture of but one size unit made possible the securing of extremely accurate performance data, simple and flexible installation arrangements and lowest possible manufacturing cost.

Fig. 1 is a cross section, illustrating the interior construction. In general the accumulator consists of a steel tank and a series of copper plate coils, arranged to form a long labyrinthian passage way for the circulating water stream. During the accumulating or charging period, thin sheets of ice are formed on each side of the plate coils under automatic control. Due to the high latent heat of fusion of water in forming ice, a large amount of *refrigeration effect* is stored compactly. During the utilization or discharge period, the ice is melted by a stream of water circulating between the cooling unit and the accumulator.

In order to control the thickness of the ice formation on each plate or baffle the refrigerant is fed independently and in progressively varying amounts to each element. Likewise the suction connection from each plate is connected to a suction header in such manner as to provide varying suction pressure (and temperatures) for each element and in a counter-progressive sequence, as related to the liquid refrigerant supply.

Performance Tests

(a) *Apparatus and Method:* A single fully charged accumulator was connected, through a water circulating line to a gas-fired water heater. The temperature of the circulating water was controlled to

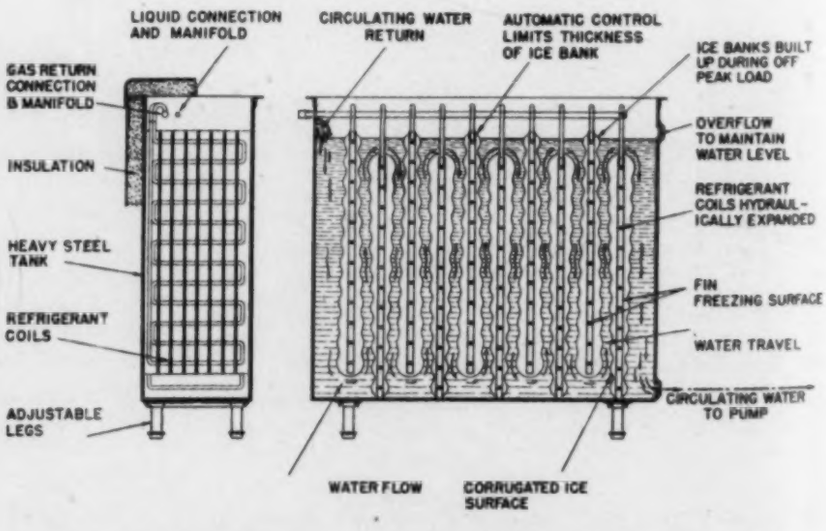

Fig. 1—Typical illustration of the general design of icy-flo accumulator for air conditioning applications

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maintain a constant temperature difference of 6 F between the inlet to and outlet from the accumulator. A large number of complete tests was conducted, for different water circulating rates; the water circulation was measured and maintained at a constant value for each test.

Because the water circulation and the difference between the inlet and outlet water temperatures were maintained at constant values for each test, the accumulator was discharged at a constant rate in Btu per minute.

	Water Circulating Rate GPM	Discharge Rate Btu per min
1	15	750
2	22.5	1125
3	30	1500
4	37.5	1875
5	45	2250
6	60	3000

The temperature of the water leaving the accumulator was measured and tabulated and the elapsed time noted for each 1 F increase of leaving water temperature.

The results of these tests were plotted for several values of continuous water flow, six of which are shown in Fig. 2.

(b) *Analysis of Test Data:* The tests show that the available cooling effect or capacity which can be obtained from a single accumulator depends upon the temperature of the cooling water required.

In air conditioning applications, cooling water temperatures of from 40 F to 50 F are required, the exact requirement being determined by the latent to total heat gain ratio.

The total available cooling effect which can be obtained from one of these particular accumulators may be determined by selecting the desired cooling water temperature, following the horizontal temperature line to the right in Fig. 2 to the intersection of the desired water circulating rate curve obtained by calculation from the peak hour heat gain, and multiplying the corresponding accumulator discharge rate in Btu per minute at this intersection by the elapsed time in minutes. This product is the net available Btu capacity of the accumulator when delivering this temperature water.

For example, assume the following:

Calculated peak hour heat gain 90,000 Btu per hour

Latent to total heat gain ratio such that 40 F cooling water is required

$$\frac{\text{90,000 Btu per hour}}{\text{8.33 lb per gal} \times \text{60 min} \times \text{6 deg temp rise circulating rate}} = \text{30 gpm water}$$

The tests described were based upon an allowable water temperature rise of 6 F. Other performance curves are available based upon tests for other allowable temperature rises.

From Fig. 2 follow 40 F horizontal water temperature line to intersection with 30 gpm water circulating curve, equivalent to a discharge rate of 1500 Btu per minute, and read 206 min of elapsed time for water delivered by the accumulator *to reach this final outlet temperature.*

1500 Btu per minute × 206 min = 309,000 Btu accumulator capacity available or 25¾ ton-hours

Assume this same calculated peak hour heat gain and latent to total heat gain ratio but with a calculated total cooling load of 50 ton-hours, (by methods hereinafter described).

A total water circulating rate of 30 gpm is still required. Two accumulators are indicated since the capacity of one accumulator when delivering 40 F cooling water is 25¾ tons-hour. The water circulating rate through each of the two accumulators would be ½ of 30 gpm or 15 gpm.

Following the 40 F water temperature line to the intersection with the 15 gpm water circulating curve, equivalent to a discharge rate of 750 Btu per minute, and read 424 min of elapsed time for water delivered by each accumulator to reach this final outlet temperature.

750 Btu per minute × 424 min = 318,000 Btu accumulator capacity of each unit or 26½ ton-hours

2 × 26½ ton-hours = 53 ton-hours total accumulator capacity (2 Units)

In like fashion any combination of peak hour demand and total ton-hours of cooling load can be met by proper choice of circulating rate and accumulator units.

Using Fig. 2 and assuming various accumulator capacity values (output values) and dividing these assumed capacity values, expressed in Btu, by the discharge rate, in Btu per minute, corresponding to various water circulation rates, the result will be expressed in elapsed time, in minutes, required for the accumulator to deliver the assumed capacities. The intersection of these elapsed time vertical lines with the water circulation curve will show the terminal temperatures of the cooling water being delivered by the accumulator while discharging these assumed capacities.

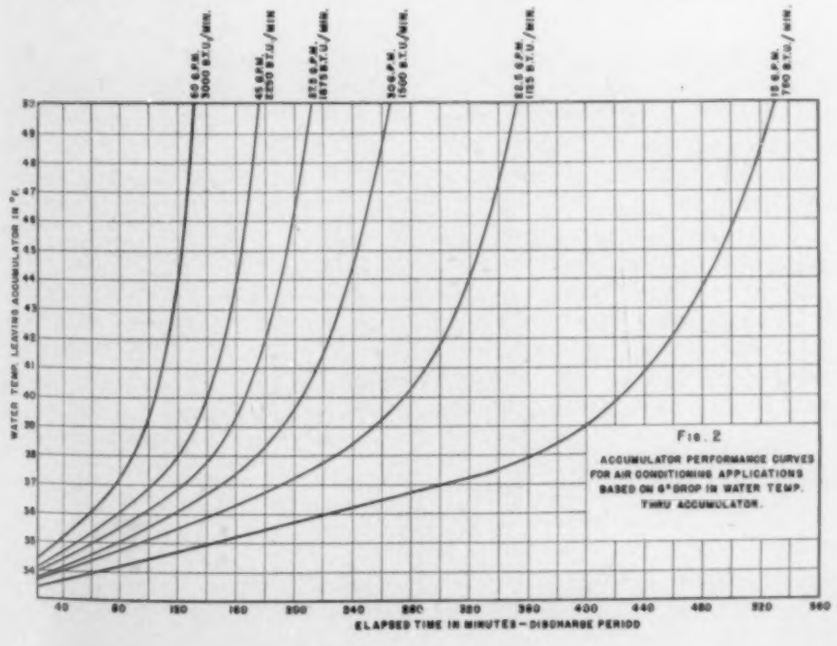
60 G.P.M. 3000 B.T.U./MIN.
45 G.P.M. 2250 B.T.U./MIN.
30 G.P.M. 1500 B.T.U./MIN.
15 G.P.M. 750 B.T.U./MIN.
WATER TEMP. LEAVING ACCUMULATOR IN °F.
ELAPSED TIME IN MINUTES—DISCHARGE PERIOD

Fig. 2 Accumulator performance curves for air conditioning applications based on 6° drop in water temp. thru accumulator.

Table 1—Capacity and Available Cooling Water Temperature One Unit Accumulator

Flow GPM		15	22.5	30	37.5	45	60	
Output		Final Water Temp deg Fahrenheit						Avg
Btu	Ton-Hours							
288,000	24	38.5	39.0	39.0	38.7	38.6	38.7	38.8
300,000	25	39.0	39.5	39.6	39.3	39.1	39.3	39.3
312,000	26	39.7	40.1	40.2	40.0	39.8	40.0	40.0
324,000	27	40.4	40.9	41.0	40.9	40.5	40.8	40.8
336,000	28	41.3	41.8	42.0	41.8	41.4	41.6	41.7
348,000	29	42.4	42.8	43.2	42.8	42.5	42.6	42.7
360,000	30	43.7	44.1	44.5	44.0	43.9	43.8	44.0
372,000	31	45.3	45.9	46.0	45.7	45.5	45.5	45.7
384,000	32	47.3	47.7	47.5	47.5	47.3	47.8	47.5
396,000	33	49.7	49.9	49.6	49.8	49.9	50.0	49.8

Table 1 has been constructed on this basis and capacity values given are for one accumulator. For multiple installation it is only necessary to multiply these capacity or output values by the number of accumulators used.

Table 1 shows that the total ton-hours output of one such accumulator is dependent only upon the temperature of the water to be delivered by the accumulator, regardless of the total water circulation required. The rate of water circulation through the accumulator determines only the rate at which its capacity is to be utilized.

If 50 F cooling water is required approximately 33 ton-hours of cooling effect is available from a single unit, whereas if 40 F cooling water is required, approximately 26 ton-hours of cooling effect would be available from a single unit.

The 33 ton-hours could be delivered in 2 hours, 12 min if 60 gpm of water circulation is used through the single unit or extended to a delivery period of 8 hours, 50 min if 15 gpm of water circulation is used.

The 26 ton-hours could be delivered in 1 hour, 44 min, if 60 gpm of water circulation is used or extended to a delivery period of 7 hours, 4 min if 15 gpm of water circulation is used.

Large cooling loads are easily handled by selecting the correct number of accumulators to provide the total storage capacity required. For example, if an application indicated the need of a total storage capacity of 300 ton-hours, with circulating cooling water of 44 F, 10 accumulators should be installed.

If the peak hour heat gain calculation indicated that the highest demand on the system would be 900,000 Btu per hour (15,000 Btu per minute) it would be necessary to circulate $\frac{15{,}000}{8.33 \times 6 \text{ deg}} = 300$ gpm, or 30 gpm per accumulator.

Charging tests have shown that a "Freon" compressor operates at an average suction refrigerant temperature of 24 F when its capacity is 10 per cent or less of the total accumulator capacity. The size of the charging compressor to be used will depend upon the hours of charging time available, before the next discharge period begins. Approximately 10,000 Btu per hour per horsepower can be delivered by the conventional water cooled "Freon" compressor unit at 24 F suction temperature.

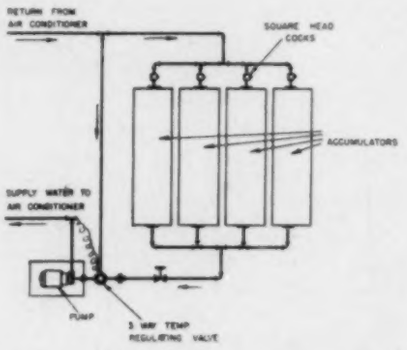

Fig. 3—Typical piping diagram of a multiple accumulator installation

Fig. 3 shows a typical piping diagram of a multiple accumulator installation. A bypass line is installed between the main supply and return water lines to and from the air conditioner through a 3-way temperature control valve, the control element of which is attached to the cold water supply line to the air conditioner. This control element is set to maintain a constant supply water temperature delivered to the air conditioner. Square head cocks are installed in the return water line to each accumulator to balance the water flow through each unit. The circulating pump is only operated during the discharge period, no water circulation being required during the charging period.

Section III—Calculation of Total Cooling Load

To properly apply a cold accumulator, it is necessary to determine the total accumulated cooling load for a given cooling period.

The various components of cooling load must be determined separately in any heat gain calculation to obtain either hourly peak demand or total operating period load.

In the determination of *hourly peak heat gain* from conduction through walls, windows, roofs, etc., multiply these various areas by unit conduction coefficients and design temperature differences between the inside and outside of the enclosure to be cooled.

The accumulated conduction cooling load would similarly be obtained if the product of these same areas and conduction coefficients were multiplied by previously determined *accumulated degree-hours* for a given cooling period. The same accumulated *degree hours* would be used to calculate the total sensible heat load from ventilation air introduced into the cooled space. In the same manner, the total cooling load due to sun effect would be determined for any given operating period if the product of exposed wall, roof or window areas and reflection and conduction factors were multiplied by previously determined *accumulated sun intensity* values.

(A) *Tables of Accumulated Degree Hours:* Fig. 4 is a reproduc-

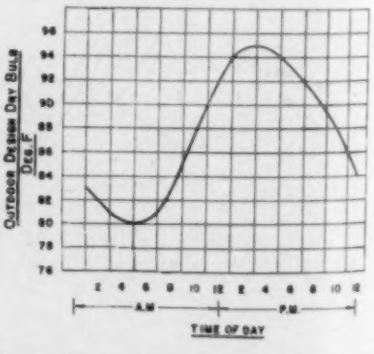

Fig. 4—Reproduction of a curve showing typical variations of outdoor temperatures for a 24-hour period

Table 2—Design Outdoor Temperatures

State	City	Design Dry-Bulb	Design Wet-Bulb	Design Mean Temperature Range
Alabama	Birmingham	95	78	15
	Mobile	95	80	15
Arizona	Phoenix	105	76	20
Arkansas	Little Rock	95	78	15
California	Los Angeles	90	70	20
	San Francisco	90	65	10
Colorado	Denver	95	64	22.5
Connecticut	New Haven	95	75	20
Delaware	Wilmington	95	78	20
District of Columbia	Washington	95	78	15
Florida	Jacksonville	95	78	15
	Tampa	94	79	12.5
Georgia	Atlanta	95	76	15
	Savannah	95	78	15
Idaho	Boise	95	65	25
Illinois	Chicago	95	75	15
	Peoria	95	76	20
Indiana	Indianapolis	95	76	20
Iowa	Des Moines	95	77	20
Kansas	Wichita	100	75	15
Kentucky	Louisville	95	76	15
Louisiana	New Orleans	95	79	12.5
	Shreveport	100	78	15
Maine	Portland	90	73	15
Maryland	Baltimore	95	78	15
Massachusetts	Boston	92	75	17.5
Michigan	Detroit	95	75	17.5
Minnesota	Minneapolis	95	75	20
	Duluth			20
Mississippi	Vicksburg	95	78	15
Missouri	Kansas City	100	76	20
	St. Louis	95	78	17.5
Montana	Helena	95	67	25
Nebraska	Lincoln	95	75	17.5
	Omaha	95	75	17.5
Nevada	Reno	95	65	30
New Hampshire	Concord	90	73	20
New Jersey	Trenton	95	78	20
New York	Albany	92	75	20
	Buffalo	93	75	15
	New York	95	75	20
New Mexico	Santa Fe	90	65	15
North Carolina	Asheville	90	75	15
	Wilmington	95	79	15
North Dakota	Bismarck	95	73	25
Ohio	Cincinnati	95	78	15
	Cleveland	95	75	15
Oklahoma	Oklahoma City	101	76	20
Oregon	Portland	90	65	20
Pennsylvania	Philadelphia	95	78	15
	Pittsburgh	95	75	20
Rhode Island	Providence	93	75	20
South Carolina	Charleston	95	80	12.5
	Greenville	95	76	15
South Dakota	Sioux Falls	95	75	20
Tennessee	Chattanooga	95	77	15
	Memphis	95	78	15
Texas	Dallas	100	78	15
	El Paso	100	69	20
	Galveston	95	80	10
	Houston	95	78	15
	San Antonio	100	78	20
Utah	Salt Lake City	92	63	25
Vermont	Burlington	90	73	20
Virginia	Norfolk	95	78	15
	Richmond	95	78	17.5
Washington	Seattle	85	65	17.5
	Spokane	90	65	17.5
West Virginia	Parkersburg	95	75	20
Wisconsin	Madison	95	75	20
	Milwaukee	95	75	20
Wyoming	Cheyenne	95	65	25

tion[1] of a curve showing typical variations of outdoor temperatures for a 24-hour period.

This temperature variation curve is based upon the mean difference between the daily maximum and daily minimum outdoor temperatures for the month of July, as explained in the text of this paper.[2] This mean difference is termed by the U. S. Weather Bureau as the *mean daily range* for July.

In the ASHVE Guide, 1941, p. 126, is tabulated outdoor *design* values of dry-bulb and wet-bulb temperatures recommended for heat gain calculations for various points in the United States. Table 2 reproduces these design dry-bulb and wet-bulb values and adds a third value entitled *design mean temperature range* for these same communities.

The design mean temperature range values given in the table, have been determined from an examination of 1940 weather reports and comparison of these values with the normal July mean temperature range given in most weather reports.

While the values given in the table are believed to be accurate, it would be advisable, before selecting a design mean temperature range, for accumulated load calculations hereinafter described, to check with

[1] A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, by F. H. Faust, L. Levine and F. O. Urban. (ASHVE Transactions, Vol. 41, 1935, p. 331.)

[2] Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan and F. O. Urban. (ASHVE Transactions, Vol. 45, 1939, p. 369.)

Table 3-A—Accumulated Degree Hours Maximum Design Dry-Bulb Temperature Differential = 12 F

July Mean Temp. Range	10 deg	12.5 deg	15 deg	17.5 deg	20 deg
5 a.m.	—	—	—	—	—
6 a.m.	2	—	—	—	—
7 a.m.	5	—	—	—	—
8 a.m.	8	2	—	—	—
9 a.m.	13	5	1	—	—
10 a.m.	19	10	4	3	1
11 a.m.	27	17	10	6	5
12 Noon	37	26	18	14	12
1 p.m.	47	37	28	23	21
2 p.m.	59	48	39	34	32
3 p.m.	71	60	51	46	44
4 p.m.	83	72	63	58	56
5 p.m.	95	84	75	70	68
6 p.m.	106	95	86	81	78
7 p.m.	116	106	95	90	87
8 p.m.	126	115	104	97	94
9 p.m.	134	123	110	103	99
10 p.m.	142	130	115	107	102
11 p.m.	148	135	119	109	103
12 Mid.	154	139	121	110	—
1 a.m.	158	142	122	—	—
2 a.m.	162	143	—	—	—
3 a.m.	165	—	—	—	—
4 a.m.	167	—	—	—	—
5 a.m.	169	—	—	—	—

Table 3-B—Accumulated Degree Hours Maximum Design Temperature Differential = 15 F

July Mean Temp. Range	10 deg	12.5 deg	15 deg	17.5 deg	20 deg
5 a.m.	—	—	—	—	—
6 a.m.	5	—	—	—	—
7 a.m.	11	5	1	—	—
8 a.m.	17	9	3	1	—
9 a.m.	25	15	7	4	1
10 a.m.	34	24	13	9	5
11 a.m.	45	34	22	17	12
12 Noon	58	46	34	28	22
1 p.m.	71	64	46	40	34
2 p.m.	86	74	61	55	48
3 p.m.	101	88	76	70	63
4 p.m.	116	104	91	85	78
5 p.m.	131	118	105	99	92
6 p.m.	145	132	119	113	106
7 p.m.	158	145	132	125	118
8 p.m.	171	157	143	136	128
9 p.m.	182	168	153	145	136
10 p.m.	194	178	161	152	142
11 p.m.	203	186	168	157	146
12 Mid.	212	193	173	160	147
1 a.m.	219	198	177	162	—
2 a.m.	226	203	180	163	—
3 a.m.	232	207	181	—	—
4 a.m.	237	210	—	—	—
5 a.m.	242	—	—	—	—

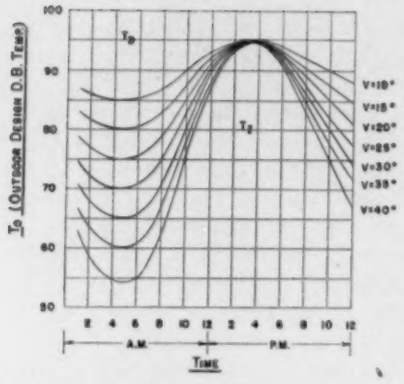

Fig. 5 — Typical outdoor temperature variation curve for seven values of July mean temperature ranges

the nearest weather bureau to obtain the normal July mean temperature range for that particular locality.

Fig. 5 is a typical outdoor temperature variation curve for seven values of July mean temperature ranges (V = 10 F, 15 F, 20 F, 25 F, 30 F, 35 F, 40 F). These curves are based upon an outdoor design dry-bulb temperature of 95 F. An indoor design dry-bulb temperature of 80 F is also indicated in Fig. 5, to show the effect of a small or large mean temperature range on the total 24-hour cooling requirement.

Reference to Table 2 shows that most design values of July mean temperature range fall between 10 F and 20 F.

In Tables 3-A, B and C, are given *accumulated degree hours* for five design values of mean temperature range (10 F, 12.5 F, 15 F, 17.5 F, 20 F). These accumulated degree hours were obtained by plotting typical outdoor temperature variation curves similar to Fig. 5 and integrating the area of the curves above assumed values of indoor design temperatures. Table 3-A is to be used for a design indoor-outdoor dry-bulb temperature difference of 12 F; Table 3-B, for a design indoor-outdoor dry-bulb temperature difference of 15 F; Table 3-C, for a design indoor-outdoor dry-bulb temperature difference of 18 F.

In general, the 12 F indoor-outdoor design temperature difference will be used where the maximum outdoor design dry-bulb temperature is from 88 F to 92 F, the 15 F value where the maximum outdoor design dry-bulb temperature is from 93 F to 97 F and the 18 F value where the maximum outdoor design dry-bulb temperature is from 98 F to 100 F.

In using these tables, the *difference* in degree hours between the start and finish of a cooling period is to be used. For example, assume a theater operating from noon until midnight (12 hours) in a community where the outdoor design dry-bulb temperature, T_o, is 92 F; indoor design dry-bulb temperature, T_i, is 80 F; and design mean temperature range, V, is 15 F. From Table 3-A, Column 3 (V = 15 F).

Accumulated degree hours at midnight	121
Accumulated degree hours at noon	18
Design degree-hours	103

Fig. 6 is a typical accumulated degree-hour curve for a design outdoor-indoor temperature difference of 15 F and for seven values of design mean temperature range. This curve graphically illustrates the data given in Table 3-B together with additional values for other design mean temperature ranges.

(B) *Tables of Accumulated Sun Intensity Values:* In the previously mentioned paper[3] are given sun effect coefficients at various latitudes for walls, roofs, windows and skylights facing several directions. The ASHVE Guide also presents similar tables for solar radiation factors expressed in Btu per square foot per hour. As explained in the text of Mr. Faust's (and others) paper, the wall sun effect intensity factors in the tables are to be multiplied by, first a reflection factor (0.4, 0.7 or 0.9) depending upon whether the wall under consideration is light colored, medium colored, or a black surface and secondly, a factor related to the conduction heat transmission coefficient of the wall or roof under consideration.

The tables applying to glass surfaces, such as windows or skylights, are to be used directly, multiplying such factors by the window or skylight areas to obtain heat transmission to an enclosure due to sun effect. Shading factors are then applied to obtain the percentage of sun effect heat transmission to the enclosure through window or skylight areas.

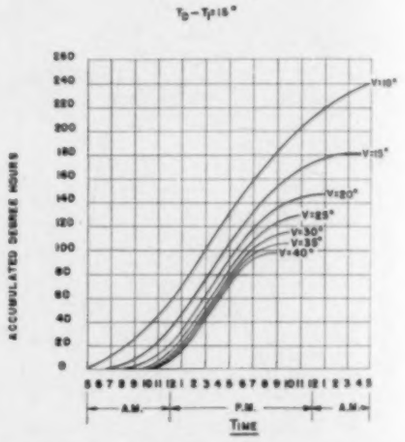

Fig. 6—Typical accumulated degree-hour curve for a design mean temperature range

In Tables 4-A, B, C and D, the tables from the ASHVE Transactions,[4] have been used for north latitudes of 30 deg, 35 deg, 40 deg, and 45 deg to obtain multipliers for exposed wall and glass areas to obtain both instantaneous and accumulated heat gain due to sun effect.

The wall sun effect factors are based upon a wall conduction coefficient, U_w, of 0.34 and a reflection factor for a medium colored wall, 0.7. The roof sun effect coefficients are based upon a roof conduction coefficient, U_r, of 0.2 and a reflection factor of a black surface, 0.9. Correction factors for other conduc-

Table 3-C—Accumulated Degree Hours Maximum Design Temperature Differential = 18 F

July Mean Temp. Range	10 deg	12.5 deg	15 deg	17.5 deg	20 deg
5 a.m.	—	—	—	—	—
6 a.m.	8	6	4	2	—
7 a.m.	17	13	8	4	—
8 a.m.	26	20	13	7	1
9 a.m.	37	29	20	13	5
10 a.m.	49	39	29	20	11
11 a.m.	63	52	41	31	21
12 Noon	79	67	55	45	34
1 p.m.	95	83	71	60	49
2 p.m.	113	101	88	77	66
3 p.m.	131	129	106	95	84
4 p.m.	149	137	124	113	102
5 p.m.	166	152	137	129	120
6 p.m.	184	169	154	145	136
7 p.m.	200	185	169	160	151
8 p.m.	216	200	184	174	164
9 p.m.	230	213	196	186	175
10 p.m.	244	226	208	196	184
11 p.m.	256	237	217	204	191
12 Mid.	268	247	226	211	196
1 a.m.	278	255	232	216	199
2 a.m.	288	263	237	219	200
3 a.m.	296	269	242	221	—
4 a.m.	305	276	246	223	—
5 a.m.	313	281	249	—	—

[3]Loc. Cit. Note 1 (p. 327).

[4]Loc. Cit. Note 1 (pp. 339, 340).

Table 4-A—30 deg North Latitude Wall and Roof Sun Effect Factor R_w and R_r[a]

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Roof	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
6 a.m.	7.0	3.4	7.6	3.6	3.6	1.7	—	—	.....	.....	.....	.....	.....	.....	1.1	0.5
7 a.m.	9.2	11.5	10.9	12.9	6.2	6.7	—	—	.....	.....	.....	.....	.....	.....	3.4	2.6
8 a.m.	9.0	20.7	11.8	24.4	7.6	13.7	—	—	.....	.....	.....	.....	.....	.....	6.4	7.4
9 a.m.	7.0	28.6	10.4	35.6	7.9	21.6	0.6	0.3	.....	.....	.....	.....	.....	.....	8.9	15.1
10 a.m.	4.2	34.2	7.6	44.2	6.7	28.6	1.7	1.4	.....	.....	.....	.....	.....	.....	10.8	24.9
11 a.m.	1.1	37.0	3.9	50.1	4.8	34.5	2.5	3.7	—	—	.....	.....	.....	.....	12.1	36.2
12 N.	....	37.5	.....	52.1	2.0	37.8	2.8	6.5	2.0	1.1	—	—	—	—	12.5	48.5
1 p.m.	.....	.....	.....	.....	.....	38.7	2.5	9.3	4.8	4.8	3.9	1.7	1.1	0.6	12.1	60.7
2 p.m.	.....	.....	.....	.....	.....	.....	1.7	11.2	6.7	10.6	7.6	7.3	4.2	3.4	10.8	72.0
3 p.m.	.....	.....	.....	.....	.....	.....	0.6	12.3	7.9	17.9	10.4	21.9	7.0	9.0	8.9	81.7
4 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	7.6	25.8	11.8	27.5	9.0	16.8	6.4	89.3
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	6.1	32.8	10.9	38.7	9.3	26.1	3.4	93.8
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	3.7	37.5	7.6	47.9	7.0	34.2	1.1	96.2
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	—	38.6	—	52.2	—	37.7	—	97.5

[a] R_w factors are for wall conduction U-value = 0.34
R_r factors are for roof conduction U-value = 0.2

For other *wall* U-values multiply R_w factors by $\frac{U}{0.34}$

For other *roof* U-values multiply R_r factors by $\frac{U}{0.2}$

R_w factors are given for medium colored walls
For light colored walls—multiply R_w factors by 0.572
For very dark walls—multiply R_w factors by 1.285

Table 4-B—35 deg North Latitude Wall and Roof Sun Effect Factor R_w and R_r[a]

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Roof	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
6 a.m.	7.6	3.9	8.1	4.2	3.9	2.0	—	—	—	—	.....	.....	.....	.....	1.3	.....
7 a.m.	9.0	12.3	11.2	13.7	6.7	7.3	—	—	—	—	.....	.....	.....	.....	3.7	3.0
8 a.m.	8.4	21.0	11.8	25.4	8.1	14.6	—	—	—	—	.....	.....	.....	.....	6.4	8.6
9 a.m.	6.2	38.2	10.3	36.6	8.4	23.0	1.7	1.1	—	—	.....	.....	.....	.....	8.7	15.3
10 a.m.	3.4	32.7	7.6	45.6	7.6	31.1	3.1	3.7	—	—	.....	.....	.....	.....	14.0	25.0
11 a.m.	—	34.4	3.9	51.2	5.6	37.8	3.9	7.0	—	—	.....	.....	.....	.....	15.7	36.5
12 N.	....	....	—	53.1	3.1	42.3	4.2	11.2	3.1	1.7	—	—	.....	.....	16.2	48.5
1 p.m.	.....	.....	.....	.....	—	43.4	3.9	15.1	5.6	6.2	3.9	2.2	—	—	15.7	60.5
2 p.m.	.....	.....	.....	.....	.....	.....	3.1	18.5	7.6	12.9	7.6	8.1	3.4	1.7	14.0	71.8
3 p.m.	.....	.....	.....	.....	.....	.....	1.7	20.7	8.4	21.0	10.4	17.4	6.2	6.5	11.5	81.2
4 p.m.	.....	.....	.....	.....	.....	.....	.....	21.6	8.1	29.4	11.8	28.6	8.4	13.7	8.4	88.8
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	6.7	37.0	11.2	40.1	9.0	22.4	4.8	94.0
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	3.9	42.3	8.1	49.5	7.6	30.8	1.7	96.5
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	—	44.2	—	54.1	—	34.4	—	97.1

[a] R_w factors are for wall conduction U-value = 0.34
R_r factors are for roof conduction U-value = 0.2

For other *wall* U-values multiply R_w factors by $\frac{U}{0.34}$

For other *roof* U-values multiply R_r factors by $\frac{U}{0.2}$

R_w factors are given for medium colored walls
For light colored walls—multiply R_w factors by 0.572
For very dark walls—multiply R_w factors by 1.285

Window Sun Effect Factor R_g

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Horizontal	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
6 a,m,	115	55	125	60	50	25	.....	.....	.....	.....	.....	.....	.....	.....	10	5
7 a.m.	150	190	180	215	95	95	.....	.....	.....	.....	.....	.....	.....	.....	60	40
8 a.m.	140	340	190	400	115	200	.....	.....	.....	.....	.....	.....	.....	.....	125	135
9 a.m.	95	455	165	575	110	315	.....	.....	.....	.....	.....	.....	.....	.....	190	295
10 a.m.	45	525	110	715	90	415	10	5	.....	.....	.....	.....	.....	.....	235	505
11 a.m.	....	550	40	790	55	490	20	20	.....	.....	.....	.....	.....	.....	260	755
12 N.	.....	.....	.....	810	10	520	20	40	10	5	.....	.....	.....	.....	270	1020
1 p.m.	.....	.....	.....	.....	.....	525	20	60	55	35	40	20	.....	.....	260	1285
2 p.m.	.....	.....	.....	.....	.....	.....	10	75	90	210	110	100	45	20	235	1535
3 p.m.	.....	.....	.....	.....	.....	.....	.....	80	110	310	165	240	95	90	190	1745
4 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	115	425	190	420	140	210	125	1900
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	95	530	180	605	150	355	60	1995
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	50	605	125	755	115	490	10	2030
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	.....	630	.....	820	.....	550	.....	2035

R_g factors are for unshaded windows
If awnings are used multiply R_g factors by 0.25
If Venetian blinds are used multiply R_g factors by 0.5

Window Sun Effect Factor R_g

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Horizontal	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
6 a.m.	120	60	135	70	55	30	.....	.....	.....	.....	.....	.....	.....	.....	15	10
7 a.m.	145	190	185	230	100	110	.....	.....	.....	.....	.....	.....	.....	.....	65	55
8 a.m.	130	330	190	420	125	220	.....	.....	.....	.....	.....	.....	.....	.....	130	155
9 a.m.	85	440	165	595	125	345	.....	.....	.....	.....	.....	.....	.....	.....	185	315
10 a.m.	30	500	110	735	105	460	25	15	.....	.....	.....	.....	.....	.....	230	525
11 a.m.	....	515	40	810	70	550	40	50	.....	.....	.....	.....	.....	.....	255	770
12 N.	.....	.....	.....	830	25	595	45	95	25	15	.....	.....	.....	.....	265	1030
1 p.m.	.....	.....	.....	.....	.....	605	40	140	70	65	40	20	.....	.....	255	1290
2 p.m.	.....	.....	.....	.....	.....	.....	25	170	105	150	110	95	30	15	230	1535
3 p.m.	.....	.....	.....	.....	.....	.....	.....	185	125	265	165	235	85	75	185	1745
4 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	125	390	190	415	130	185	130	1905
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	100	505	185	605	145	320	65	2005
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	55	585	135	765	120	455	15	2045
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	.....	615	.....	835	.....	515	.....	2055

R_g factors are for unshaded windows
If awnings are used multiply R_g factors by 0.25
If Venetian blinds are used multiply R_g factors by 0.5

Table 4-C—40 deg North Latitude Wall and Roof Sun Effect Factor R_w and R_r[a]

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Roof	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
6 a.m.	7.9	3.9	6.7	4.5	4.5	2.3	—	—	—	—	—	—	.....	.....	3.0	0.8
7 a.m.	9.0	12.3	11.2	14.3	7.0	7.9	—	—	—	—	—	—	.....	.....	3.8	3.6
8 a.m.	7.9	20.7	11.8	26.0	8.7	15.7	0.6	0.3	—	—	—	—	.....	.....	6.3	8.7
9 a.m.	5.6	27.4	10.4	36.7	9.0	24.6	2.5	2.0	—	—	—	—	.....	.....	8.5	16.1
10 a.m.	2.2	31.4	7.6	45.6	8.4	33.3	4.2	5.3	—	—	—	—	.....	.....	10.2	25.4
11 a.m.	—	32.5	3.9	51.2	6.5	40.6	5.3	10.1	0.9	0.6	—	—	.....	.....	11.4	36.2
12 N.	....	....	—	53.2	3.9	45.6	5.6	15.7	3.9	2.8	—	—	.....	.....	11.9	47.8
1 p.m.	....	....	.....	.....	0.9	48.1	5.3	21.3	6.5	7.9	3.9	2.0	—	—	11.4	59.5
2 p.m.	....	....	.....	.....	—	48.5	4.2	26.0	8.4	15.4	7.3	7.9	2.2	1.1	10.2	70.2
3 p.m.	....	....	.....	.....	.....	.....	2.5	29.4	9.0	24.1	10.4	16.8	5.6	5.1	8.5	79.5
4 p.m.	....	....	.....	.....	.....	.....	0.6	30.8	8.7	33.0	11.8	27.7	7.9	11.8	6.5	86.9
5 p.m.	....	....	.....	.....	.....	.....	—	31.0	7.0	40.8	11.2	39.2	9.0	20.2	3.8	92.0
6 p.m.	....	....	.....	.....	.....	.....	.....	.....	4.5	46.7	8.7	49.0	7.9	28.6	3.0	94.8
7 p.m.	....	....	.....	.....	.....	.....	.....	.....	—	49.0	—	53.2	—	32.8	—	95.6

[a] R_w factors are for wall conduction U-value = 0.34
R_r factors are for roof conduction U-value = 0.2

For other *wall* U-values multiply R_w factors by $\frac{U}{0.34}$

For other *roof* U-values multiply R_r factors by $\frac{U}{0.2}$

R_w factors are given for medium colored walls
For light colored walls—multiply R_w factors by 0.572
For very dark walls—multiply R_w factors by 1.285

Window Sun Effect Factor R_g

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Horizontal	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
6 a.m.	130	60	145	75	65	35	.....	.....	.....	.....	.....	.....	.....	.....	20	10
7 a.m.	145	200	185	240	105	120	.....	.....	.....	.....	.....	.....	.....	.....	70	60
8 a.m.	120	335	190	430	135	240	.....	.....	.....	.....	.....	.....	.....	.....	130	165
9 a.m.	70	430	165	610	140	380	20	10	.....	.....	.....	.....	.....	.....	180	325
10 a.m.	15	475	110	745	125	515	45	45	.....	.....	.....	.....	.....	.....	220	525
11 a.m.	....	480	40	815	90	620	65	100	.....	.....	.....	.....	.....	.....	250	760
12 N.	....	....	....	835	40	685	70	170	40	20	.....	.....	.....	.....	255	1015
1 p.m.	....	....	.....	.....	....	705	65	240	90	85	40	20	.....	.....	250	1270
2 p.m.	....	....	.....	.....	.....	.....	45	290	125	195	110	95	15	10	220	1505
3 p.m.	....	....	.....	.....	.....	.....	20	320	140	325	165	235	70	55	180	1705
4 p.m.	....	....	.....	.....	.....	.....	.....	330	135	465	190	415	120	155	130	1855
5 p.m.	....	....	.....	.....	.....	.....	.....	.....	105	585	185	605	145	290	70	1955
6 p.m.	....	....	.....	.....	.....	.....	.....	.....	65	670	140	765	130	430	20	2000
7 p.m.	....	....	.....	.....	.....	.....	.....	.....	.....	700	.....	840	.....	490	.....	2010

R_g factors are for unshaded windows
If awnings are used multiply R_g factors by 0.25
If Venetian blinds are used multiply R_g factors by 0.5

Table 4-D—45 deg North Latitude Roof and Wall Sun Effect Factor R_w and R_r[a]

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Roof	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
5 a.m.	3.9	2.0	3.9	2.0	1.4	0.9	—	—	—	—	—	—	—	—	—	—
6 a.m.	8.1	8.1	9.3	6.7	4.8	3.9	—	—	—	—	—	—	—	—	1.8	1.3
7 a.m.	8.7	16.5	11.5	19.0	7.3	10.2	—	—	—	—	—	—	—	—	4.0	4.0
8 a.m.	7.3	24.6	11.8	30.8	9.2	18.5	1.4	0.9	—	—	—	—	—	—	6.1	9.1
9 a.m.	4.5	30.8	10.2	41.7	9.8	28.0	3.6	3.4	—	—	—	—	—	—	8.2	16.3
10 a.m.	1.4	33.9	7.6	50.4	9.0	36.7	5.3	7.8	—	—	—	—	—	—	9.7	25.4
11 a.m.	—	34.7	3.9	56.0	7.3	44.8	6.4	13.7	1.7	0.9	—	—	—	—	11.1	35.6
12 N.	....	....	.....	58.0	5.1	51.0	7.0	20.5	5.0	4.2	—	—	—	—	11.3	46.9
1 p.m.	....	....	.....	.....	1.7	54.6	6.4	27.1	7.3	10.4	—	—	—	—	11.1	57.9
2 p.m.	....	....	.....	.....	—	55.5	5.3	33.0	9.0	18.5	7.6	7.6	1.4	0.9	9.7	68.2
3 p.m.	....	....	.....	.....	.....	.....	3.6	37.5	9.8	28.0	10.2	16.5	4.8	3.9	8.2	77.4
4 p.m.	....	....	.....	.....	.....	.....	1.4	40.0	9.3	37.5	11.8	27.4	7.3	10.2	6.1	84.5
5 p.m.	....	....	.....	.....	.....	.....	—	40.8	7.3	45.8	11.5	39.2	8.7	18.2	4.0	89.5
6 p.m.	....	....	.....	.....	.....	.....	.....	.....	4.8	52.1	9.3	49.5	8.1	26.6	1.8	92.9
7 p.m.	....	....	.....	.....	.....	.....	.....	.....	1.4	55.1	3.9	56.2	3.9	32.5	—	93.3
8 p.m.	....	....	.....	.....	.....	.....	.....	.....	—	55.7	—	58.2	—	34.4	—	—

[a] R_w factors are for wall conduction U-value = 0.34
R_r factors are for roof conduction U-value = 0.2

For other *wall* U-values multiply R_w factors by $\frac{U}{0.34}$

For other *roof* U-values multiply R_r factors by $\frac{U}{0.2}$

R_w factors are given for medium colored walls
For light colored walls—multiply R_w factors by 0.572
For very dark surfaces (roofs, etc.)—multiply above factors by 1.285

Window Sun Effect Factor R_g

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Horizontal	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
5 a.m.	65	40	65	35	20	10	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
6 a.m.	135	140	150	145	70	55	.....	.....	.....	.....	.....	.....	.....	.....	25	15
7 a.m.	140	280	190	315	115	150	.....	.....	.....	.....	.....	.....	.....	.....	70	65
8 a.m.	110	405	190	505	145	280	.....	.....	.....	.....	.....	.....	.....	.....	125	165
9 a.m.	55	485	165	680	150	430	35	15	.....	.....	.....	.....	.....	.....	175	315
10 a.m.	....	515	110	815	140	575	65	65	.....	.....	.....	.....	.....	.....	215	510
11 a.m.	....	....	40	890	105	700	85	140	10	5	.....	.....	.....	.....	240	740
12 N.	....	....	.....	910	55	780	95	230	55	35	.....	.....	.....	.....	245	985
1 p.m.	....	....	.....	.....	10	810	85	320	105	115	40	20	.....	.....	240	1230
2 p.m.	....	....	.....	.....	.....	815	65	395	140	235	110	95	.....	.....	215	1460
3 p.m.	....	....	.....	.....	.....	.....	35	445	150	380	165	235	55	25	175	1655
4 p.m.	....	....	.....	.....	.....	.....	.....	465	145	530	190	415	110	110	125	1805
5 p.m.	....	....	.....	.....	.....	.....	.....	.....	115	660	190	605	140	235	70	1905
6 p.m.	....	....	.....	.....	.....	.....	.....	.....	70	750	160	775	135	375	25	1955
7 p.m.	....	....	.....	.....	.....	.....	.....	.....	20	800	65	885	65	475	.....	1963
8 p.m.	....	....	.....	.....	.....	.....	.....	.....	.....	810	.....	920	.....	505	.....	.....

R_g factors are for unshaded windows
If awnings are used multiply R_g factors by 0.25
If Venetian blinds are used multiply R_g factors by 0.5

tion coefficients and reflection factors are given in the tables.

The instantaneous values, for use in calculating peak heat gain are given in one column, and accumulated values in another column. The accumulated values were obtained by plotting instantaneous values and integrating the area under these curves.

In using the tables to obtain the accumulated load due to sun effect, the difference in values between the start and finish of a cooling period is to be used. For example, assume a church with a morning service beginning at 10:30 a.m. and completed at 12:00 noon, the location of the church being 30 deg north latitude and with East, South, West and Roof exposures. Select accumulated sun effect factors as follows:

Exposure	East	South	West	Roof
Wall and Roof Factors				
Accumulated R_W (or R_T) Noon	52.2	6.4	None	48.5
Accumulated R_W (or R_T) 10:30 a.m. (By Interpolation)	47.3	2.1	None	30.6
Design R_W (or R_T)	4.9	4.3	None	17.9
Window Factor				
Accumulated R_S Noon	810	40	None	1020
Accumulated R_S 10:30 a.m. (By Interpolation)	752	12	None	630
Design R_S	58	28	None	390

It should be borne in mind that in calculating the *peak hour* heat gain from sun effect, only such exposures are considered which have the sun shining on them at the particular time the *peak hour* load is determined, whereas, all outdoor exposures must be considered in calculating the *accumulated total* heat gain.

Fig. 7 graphically illustrates the accumulated action of the sun shining on various exposed surfaces throughout typical July days in a location of 30 deg North Latitude.

No account is taken in the table values of the effect of heat storage in the walls. There is, of course, a definite time lag in the delivery of outdoor heat to an enclosure due to heat storage in the structure. It is essential that the design engineer take into account the effect of this stored heat when calculating the total accumulated heat gain. The value to be set up to precool the structure and to be added to the total load calculation is left to the judgment of the design engineer. In general, it will fall between two and three times the calculated peak hour conduction and sun effect load; however, type of construction, insulation used and exposure will cause a variation in the amount to be added to the net calculated load.

In this connection, it may be added that *negative conduction* and radiation will result in some cases, where the outdoor temperatures drop below indoor design values after the sun goes down. This is effectively shown in Fig. 5, particularly where design mean temperature ranges are high. Such negative loads could be calculated and deducted from the total previously determined accumulated cooling load.

In calculating the total sensible cooling period load from occupancy, light and appliances, and latent cooling period load from occupancy, ventilation air and appliances it may be assumed that these loads are constant throughout the cooling period under consideration. It is therefore only necessary to multiply the *peak hour* value of these components of heat gain by the hours of the cooling cycle, to obtain their contribution to the total cooling load.

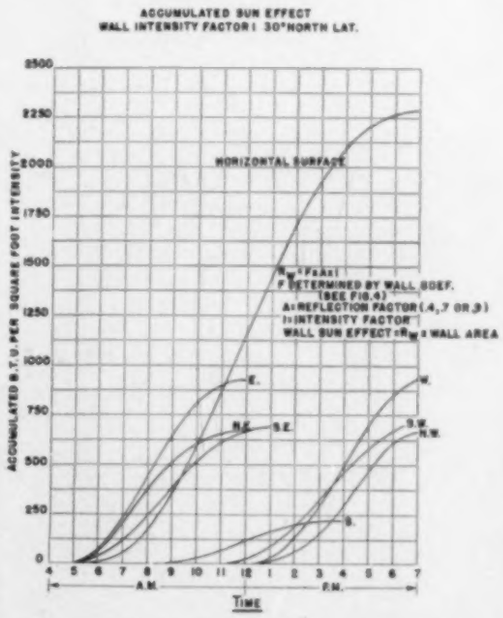

Fig. 7—Graphic illustration of the accumulated action of the sun shining on various exposed surfaces throughout typical July days—30 deg North Latitude

Peak hour heat gains are calculated in Btu per hour and similarly, using the table values of accumulated degree-hours and accumulated sun intensity values, the total cooling load will be expressed in Btu. The peak hour heat gain may be expressed in tons of cooling demand and the total load in ton-hours by dividing the calculated results by 12,000.

(C) *Selection of Required Accumulator Capacity:*

Example: Industrial Cafeteria

Total cooling load—960,000 Btu = 80 ton-hours.

Peak hour cooling demand—288,000 Btu per hour = 24 tons.

Cooling period ..... 5 hours

Non-operating period ..... 19 hours

Total ..... 24 hours

46 F cooling water required to meet latent to total heat ratio.

Minimum charging capacity

$$\frac{80 \text{ ton-hours}}{24 \text{ hours}} = 3.33 \text{ tons}$$

Approximate capacity "Freon" compressors at 24 F suction temperature 75 F cooling water.

3 hp — 2.2 tons

5 hp — 4.4 tons

Select 5 hp "Freon" compressor.

(a) *Assume Charging Compressor Operating During Cooling Period*

5 hours × 4.4 tons = 22 ton-hours available from compressor operation.

80 ton-hours — 22 ton-hours = 58 ton-hours of accumulator capacity required.

Referring to Fig. 8, two accumulators will have a capacity of 60 ton-hours when delivering 46 F cooling water (intersection of 60 ton-hour and 46 F water lines).

Following the 60 ton-hour line to the right to the 4.4 ton per hour charging line, 13.6 hours of charging time will be required out of 19 hours available, leaving 5.4 hours shut down period (intersection of 4.4 ton charging line with 13.6 interpolated hour line).

(b) *Assume Charging Compressor Not Operating During Cooling Period*

80 ton-hours accumulator capacity required.

Referring to Fig. 8, three accumulators will have a capacity of 90 ton-hours when delivering 46 F cooling water (intersection of 90 ton-hour and 46 F water lines).

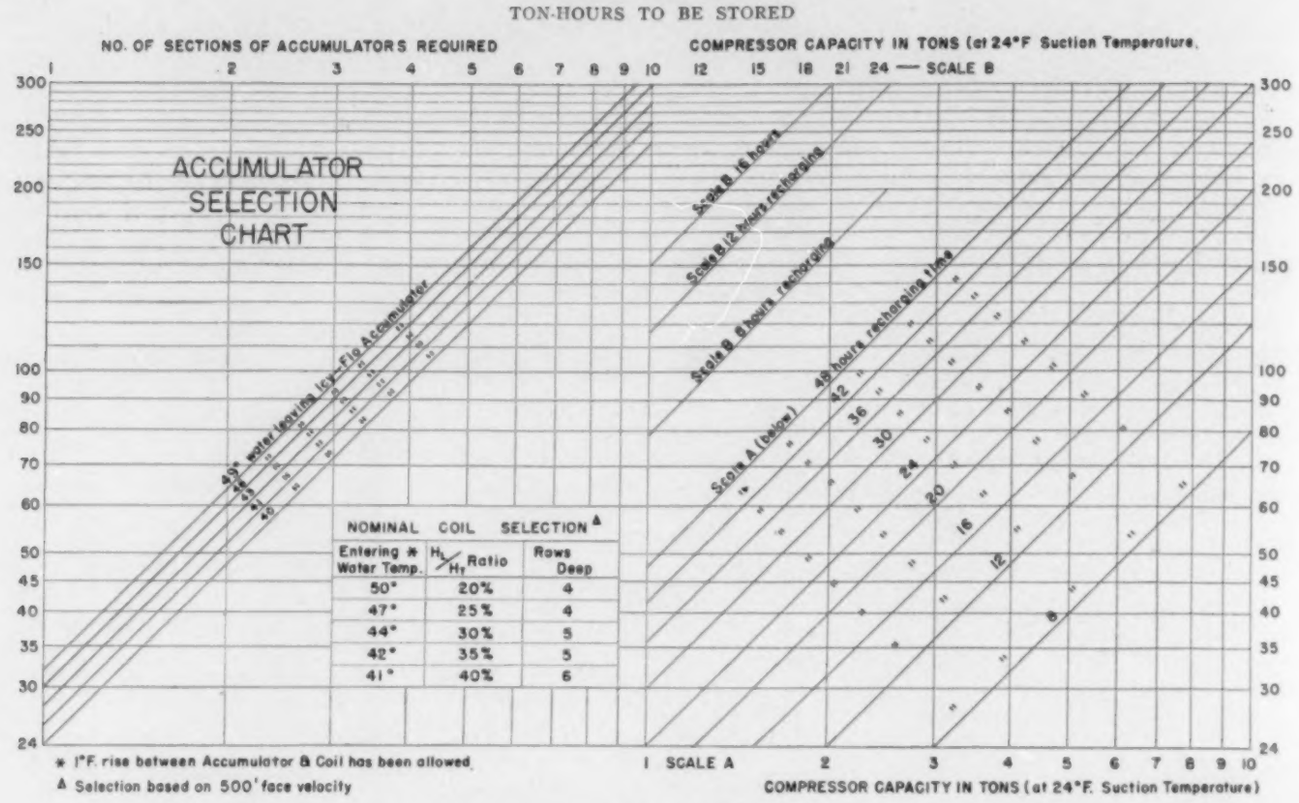
TON-HOURS TO BE STORED
NO. OF SECTIONS OF ACCUMULATORS REQUIRED
COMPRESSOR CAPACITY IN TONS (at 24°F Suction Temperature.
ACCUMULATOR SELECTION CHART
Scale B 16 hours
Scale B 12 hours recharging
Scale B 8 hours recharging
Scale A (below) 48 hours recharging time
NOMINAL COIL SELECTION Δ
Entering * Water Temp. | H_L/H_T Ratio | Rows Deep
50° | 20% | 4
47° | 25% | 4
44° | 30% | 5
42° | 35% | 5
41° | 40% | 6
* 1°F. rise between Accumulator & Coil has been allowed.
Δ Selection based on 500' face velocity
SCALE A
COMPRESSOR CAPACITY IN TONS (at 24°F. Suction Temperature)

Fig. 8—Accumulator selection chart

Following the 80 ton-hour storage capacity line to the right, to the 4.4 ton per hour charging line, 18.2 hours of charging time will be required out of 19 hours available. Note that if direct expansion cooling were to be used a 24 ton compressor would have to be installed to meet the peak hour demand, even though this compressor could be operated at a relatively higher suction temperature.

With the accumulators, the load demand, both peak hour and total, can be met with a 4.4 ton compressor.

Section IV—Conclusions

While the cold accumulator is particularly adapted to the handling of heavy air conditioning loads of short duration, this study of actual degree-hours of typical cooling loads for a month such as July indicates that the wide peaks and valleys in a resulting curve made up of hourly transmission, solar heat gain, ventilation load as well as variations in occupancy, make it possible, by the use of a suitable accumulator unit, to reduce the size of the compressor equipment materially, with resulting economies in first cost, operating expense and upkeep.

Further, the use of these new refrigeration *storage batteries* make practical the development of *off-peak* power loads which will reduce the cost of electric power to the owners of air conditioned buildings while making summer cooling loads more desirable to the electric utility company.

Since periods of *peak demand* on most utility systems rarely extend over more than four hours, it is simple to arrange control circuits so that the entire recharging of the accumulator cycle takes place *off-peak*, and particularly so in the case of hotel banquet halls, churches, auditoriums, mortuaries, and noonday cafeterias. In addition many office building and retail store applications will be found where this feature can be utilized to reduce electric power costs by leveling out the individual *demand* curves.

Extreme flexibility in handling a number of air conditioning zones composing one system can be secured by use of the accumulator—chilled water circulating method.

RADIATOR STANDARD

The Standing Committee in charge of reviewing and revising Simplified Practice Recommendation R174-40, large-tube cast-iron radiators, has approved a revision of the recommendation, and the Division of Simplified Practice of the *National Bureau of Standards* has mailed copies to all interests for consideration and approval.

The original draft of this recommendation, which was promulgated as of March 1, 1940, established a simplified schedule of 17 recommended stock sizes of large tube radiators out of a total of 33 sizes in production at that time.

The revision now proposed would further reduce the list of stock items to 13, by eliminating the following: (1) 4-tube, 2¼ ft² rating; 20-in. height; (2) 6-tube, 3 ft² rating, 20-in. height; (3) 7-tube, 3 ft² rating, 17-in. height, and (4) 7-tube, 4¾ ft² rating, 26-in. height.

Subject to general approval, the revised recommendation will become effective for new production on October 1, 1941, and will remain effect until it is again revised by the Standing Committee of the industry.

Mimeographed copies of the proposed revision may be obtained without charge from the Division of Simplified Practice, *National Bureau of Standards*, Washington, D. C.

Performance Characteristics of a Forced Draft, Counterflow Spray Cooling Tower

By Howard H. Niederman*, E. D. Howe**, John P. Longwell***, Berkeley, Calif.
Ralph A. Seban†, Santa Clara, Calif., and L. M. K. Boelter††, Berkeley, Calif.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of California

In small spray cooling towers, in contrast to packed cooling towers, the water distribution across any section varies widely. This distribution is a function of the type of spray used, the water pressure, water velocity, water temperature, air velocity and pressure, and of the tower construction. In general the water pressures and velocities for any given rate of flow are fixed by the type of spray, while the air velocities and pressures for any given rate of flow are fixed by the tower design and partially by the characteristics of the water spray.

There are many types of spray nozzles on the market today, but the manufacturers thereof do not have data available which will yield drop sizes and distributions. Without adequate data it is extremely difficult to predict what the performance of any given installation will be. In the present work several different types of sprays were used and their over-all characteristics noted.

Generally speaking the performance of cooling towers is expressed by means of the rate coefficients for thermal convection, mass transfer, and enthalpy transfer:

$$Gs(t_1 - t_2) = f_c a\, V(T_1 - t)_{lm} \quad \ldots\ldots(1)$$

$$G(H_1 - H_2) = f_d a\, V(H_1 - H)_{lm} \quad \ldots\ldots(2)$$

$$G(h_1 - h_2) = f_h a\, V(h_1 - h)_{lm} \quad \ldots\ldots(3)$$

Equations (1) and (2) are the integrated forms of the differential equations which are derived in reference (16),[1] and which have been presented in the paper of London, Mason, and Boelter (8). Equation (3) has been derived in the same paper, and further data on the use of the equation have been given in the recent paper of London, Nottage, and Boelter (9). The over-all conductances, $f_c a$, $f_d a$, and $f_h a$ are products of an over-all conductance between the main body of water and the main body of air, and the transfer area per unit volume of the exchanger. The main body may be defined as that part of the substance existing at a temperature indicative of the energy content of all of that substance. The use of logarithmic mean potentials in Equations (1), (2) and (3) is questionable. The logarithmic mean is only correct in those cases for which the operating line and the equilibrium line are straight over the region in question. It is doubtful whether this condition ever obtains in a spray tower, since flow characteristics change rapidly with height. The results, consequently, may only be compared accurately with dynamically similar towers, the performances of which are calculated on the same basis. Any comparison with packed towers or wetted wall towers in which the logarithmic mean applies (under the conditions specified) can only be approximate. However, for want of a better, simply computed potential the results presented in this paper are calculated on the basis of the logarithmic mean.

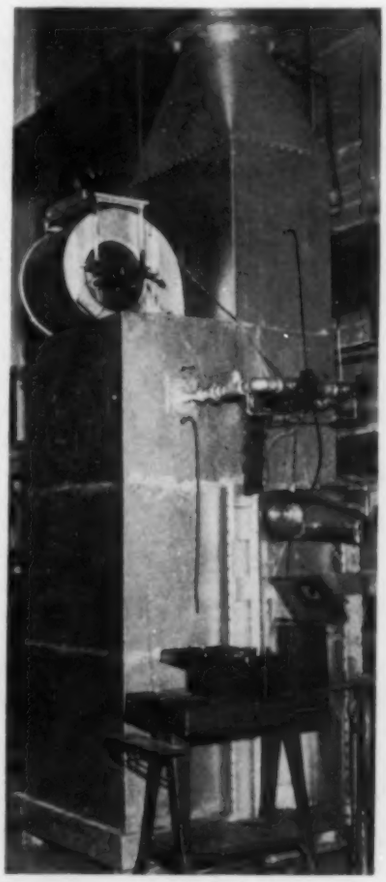

Fig. 1—View of spray cooling tower showing thermocouples, potentiometer, and air measuring devices

In order to determine whether calculation of the conductances in terms of the number of transfer units (NTU) would yield a different result from that using the logarithmic mean potential, several computations were made. The conductances obtained in the former manner agreed within 2 per cent of those obtained by the latter method.

Description of Apparatus

The tests were made on a forced draft, spray cooling tower illustrated in Fig. 1 which was divided into two vertical sections, one section providing parallel flow and the other counterflow as shown in Fig.

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Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, San Francisco, Calif., June, 1941.

[1]Numerals in parentheses refer to Bibliography.

2. Water circulation during runs took place in a closed system, the only loss being that due to evaporation. Water was pumped from the basin of the tower through a calibrated metering orifice and thence through a heat exchanger and back into the top of the tower where it was introduced through six sprays in each section. The rate of evaporation was obtained by noting the change in level of the basin with respect to time. Measurements of water level were made by means of a point gage shown in Fig. 3.

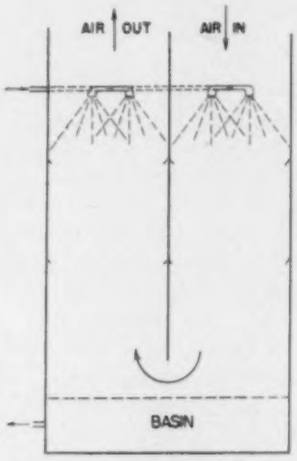


Fig. 2—Flow diagram of spray cooling tower
4 ft x 3 ft x 7 ft

Air was forced through the tower by means of a fan placed at the entrance to the parallel flow section and was metered at the outlet. The necessary temperature measurements (*i.e.*, wet- and dry-bulb temperatures of the air in and out of the tower and the temperatures of the water in and out of the tower) were taken using thermocouples placed in appropriate positions.

The piping into the tower was so arranged that the parallel and counterflow sections might be run individually or simultaneously. Inasmuch as it is difficult to determine what the driving potentials are in a parallel-counterflow system such as this, it was decided to take data on the two sections individually. The results presented here are from data taken on the counterflow side.

The construction of the tower was such that the water after leaving the sprays struck the walls within a very short distance. The drops then either coalesced and traveled down the wall in sheets, or else were deflected back into the air stream. The cooling of the water was therefore obtained from both the drop surfaces and the surfaces exposed along the walls. In order to promote turbulence (and consequent mixing) and separation from the walls, several protrusions were attached around the inside of the tower (see Fig. 2).

Since the conductances are dependent upon the transfer area per unit volume, they are inevitably dependent upon the water distribution and drop size distribution. Any conditions which will cause the transfer area, a, to increase will consequently increase the over-all conductances. The extent to which transfer area depends on drop size distribution may be seen in Table 1 which gives the surface area exposed when a given volume of liquid is distributed into drops of various sizes according to several different methods of distribution.

The given liquid volume in the case under consideration was chosen as one which might reasonably exist in 1 cu ft of the cooling tower at any moment if the water were evenly distributed throughout.

The most obvious method of obtaining greater transfer area in a spray tower is to attempt to obtain a spray distribution which will give a maximum number of fine drops throughout the length of the tower. Several different types of sprays were substituted for this purpose.

Until recently (8) the performance of cooling towers has been represented in terms of the over-all coefficient for mass transfer, $f_d a$. In order to facilitate direct comparison the results obtained are given in the same manner, although the conductances for sensible heat transfer and enthalpy transfer have been calculated and are available for comparison with new data as they are presented.

The values of conductances were computed from runs which were of sufficient duration to yield accurate measurements of the lowering of the basin level, and during which time steady state conditions held. The accuracy of the data obtained was determined from the energy and mass balances. Only those runs in which the balances checked within seven per cent were utilized.

Energy Balance

$$L\, c_p\, (T_1 - T_2) + \Delta L\, c_p\, T_2 = G\, (h_1 - h_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

The left-hand side of Equation (4) indicates the energy lost by the water, while the right-hand side indicates the energy gain of the air. The datum for water is 32 F, while that for air is 0 F. The enthalpies of the air at entrance and exit were calculated by means of Equation (5).

$$h = st + 1061H \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

The error of the energy balance was obtained by dividing the difference between the two sides of Equation (4) by the energy lost by the water. No correction was made for the thermal energy loss from the uninsulated sides of the tower. Measurements of the outside wall temperatures of the tower during several runs, taken by means of thermocouples, yielded values of the conductance (convection plus radiation) of approximately 0.8 Btu per foot^2 hour F (evaluated by means of the Heilman equation). Measurements of the over-all conductance for heat transfer through the walls to the outside air made by R. A. Seban (12) indicated a value of 1.6 Btu per foot^2 hour F. His method involved insulation of the ends of the tower and determination of the heat loss of the water through the tower with no air flowing. The conductance was then calculated directly from the equation.

$$q = f\, A\, \Delta T \quad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

The magnitude obtained by Seban, however, was at temperatures 30 to 40 F above the usual water temperatures within the tower. In either case, the thermal energy loss through the walls represents less than 2 per cent of the total energy transfer from the water to the air stream.

Fig. 3—Point gage, and outlet for water distribution pan

Table 1—Change in Surface Area of Drops

(Assume that the water is distributed equally within the tower and that there is 0.004 ft³ water per ft³ of tower volume at any moment. Assume that the drops are all spherical and that there are only five drop sizes, ½, 1, 1½, 2, 2½ mm diameter)

Drop Size		Distribution According to Equal Volumes			Distribution According to Equal Numbers of Drops			Probability Distribution of Volume			Probability Distribution of Number of Drops		
Diameter Mm.	Millifeet	Number of Drops	Volume x 10^6 ft³	Surface Area ft²	Number of Drops	Volume x 10^6 ft³	Surface Area ft²	Number of Drops	Volume x 10^6 ft³	Surface Area ft²	Number of Drops	Volume x 10^6 ft³	Surface Area ft²
0.5	1.64	347,000	800	2.93	7,720	17.8	0.06	121,800	280	1.02	3,340	7.7	0.03
1.0	3.28	43,400	800	1.46	7,720	142.2	0.26	52,100	960	1.76	11,450	211.5	0.39
1.5	4.92	12,800	800	0.98	7,720	481.0	0.59	24,390	1520	1.85	18,100	1131	1.38
2.0	6.56	5,420	800	0.73	7,720	1140	1.04	6,500	960	0.88	11,450	1687	1.54
2.5	8.20	2,780	800	0.59	7,720	2225	1.63	970	280	0.20	3,340	963	0.70
Totals		411,400	4000	6.69	38,600	4006	3.58	205,070	4000	5.71	47,680	4000	4.04
Area Per Unit Volume of Tower		=	6.69 ft²/ft³			3.58 ft²/ft³			5.71 ft²/ft³			4.04 ft²/ft³	

If the drops were all of the same average size, 1½ mm., the transfer area would be 'a' = 4.87 ft²/ft³.

Material Balance

$$\Delta L = G\,\Delta H \quad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

The left-hand side of Equation (7) gives the rate of evaporation from the basin of the tower as measured by the change in point gage level, while the right-hand side gives the change in moisture content of the air passing through the tower. The error was calculated by dividing the difference of the two sides of the equation by ΔL.

Since the water distribution within the tower is extremely important an attempt was made to determine what fraction of the water passed down the central portion of the tower. This was done by placing a collecting pan having an area equal to one-half the tower cross section in the center of the bottom of the counterflow side. The results are shown by the curves in Fig. 4 and in Fig. 5 will be found a detailed description of each one of the five nozzles. It is apparent that the best space distribution was given by Nozzle *D* and the worst by Nozzle *A*. These results are misleading, however, for they give the water distribution in only one particular portion of the tower, *i.e.*, at the bottom, and they give no indication whatsoever of comparative drop size distributions nor of the effect of air velocity on the space distribution (except for Nozzle *C*). Superficial examination of the sprays indicated that Nozzle *A* yielded the finest spray while Nozzles *D* and *E* yielded substantially coarser drops.

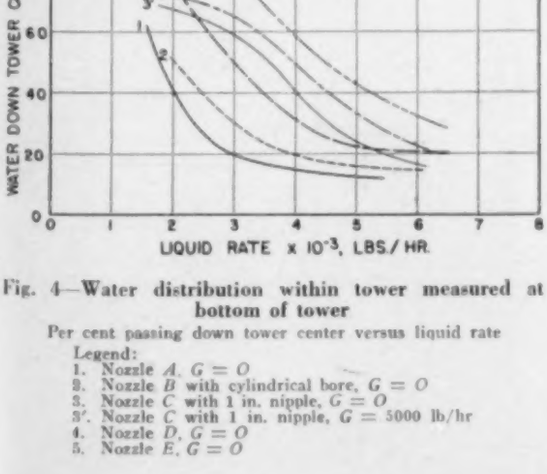


Fig. 4—Water distribution within tower measured at bottom of tower

Per cent passing down tower center versus liquid rate

Legend:
1. Nozzle *A*, $G = O$
2. Nozzle *B* with cylindrical bore, $G = O$
3. Nozzle *C* with 1 in. nipple, $G = O$
3'. Nozzle *C* with 1 in. nipple, $G = 5000$ lb/hr
4. Nozzle *D*, $G = O$
5. Nozzle *E*, $G = O$

Tower Performance

In order to determine the actual performance of the tower with the various sprays installed, steady state runs were made on the counterflow side and the conductances computed. The results are shown in Fig. 6 in which the conductance for mass transfer is plotted against the liquid rate. From these curves the actual over-all effect of the various sprays is evident. The fact that Nozzle *D* gave one of the lowest curves in contrast to the fact that it gave the best distribution is interesting for it indicates that the size of the drops is of greater importance than the space distribution of the water. The other curves of conductance versus water rate actually lie in the same relative order as the relative drop sizes, the implication being that the greatest portion of the cooling in the tower is due to the drops even though most of the water passes down the walls. In other words, most of the cooling takes place at the section where the water first enters. This conclusion is further substantiated by recent results obtained on the parallel flow side of the tower which indicate that the over-all mass conductance for parallel flow is higher than that for counterflow (17). The latter results can be explained by the correspondingly higher potential at the entrance section.

The variation of the conductance, $f_a a$, with liquid rate is probably due to the change in transfer area, a, since increased liquid velocity causes increased atomization of the water drops. It may also be due to the fact that near the nozzles cross flow occurs; that is, the flow is not truly countercurrent.

At higher magnitudes of the liquid rate, L, all the curves tend to break. In the case of the original Nozzle *A* a number of runs were made at high liquid rates to attempt to verify this change in slope. It was found that the points obtained scattered widely in comparison with points on the remainder of the curve. The probable explanation is that the amount of water passing through the nozzles has become so large that some of the water droplets coalesce immediately after emerging. The transfer area is therefore not constant but varies with the amount of coalescence of the drops. Any particular run might therefore be difficult to duplicate since the coalescence of the drops is random.

In order to determine how much cooling was done by the water falling down the walls of the tower the sprays were plugged and the water

was distributed around the walls of the tower by means of drilled pipes in such manner that the sheet surface area exposed was approximately the same as that exposed when the sprays were operating. The liquid rate and air rate were varied and it was found that the mass conductance, $f_a a$, varied approximately as $G^{0.5}$ and $L^{0.3}$ (Figs. 7 and 8)

$$f_a a = 0.033 G^{0.5} L^{0.3} \quad \ldots\ldots\ldots (8)$$

Sherwood and Holloway (15) obtained a somewhat similar equation for the absorption of ammonia in ring-packed towers

$$K_g a = 0.046 G^{0.5} L^{0.4} \quad \ldots\ldots\ldots (9)$$

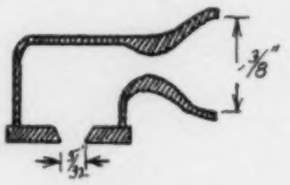
The exponent 0.5 for the air rate compares favorably with the exponent 0.48 obtained by London for mass transfer from streamlined packing sections (8). Gilliland and Sherwood report an exponent of 0.8 for wetted wall towers (3). However, in their work the flow of liquid film was viscous and consequently the relative velocity of air and water did not differ greatly from the absolute air velocity. In the present case the relative velocity was greatly different from the absolute. Relative velocities (approximate) were employed to correlate the spray tower data, and it was found that the conductance then varied approximately as the 0.8 power. It is impossible to make actual exact comparisons with results of other experimenters because of the nature of the construction of the tower, but it is believed that the results indicated are valid and that they are partially substantiated by the closeness of the comparisons given.

A complete series of runs was made using Nozzle *A* and varying the liquid and air rates. The results (Figs. 7 and 8) indicated that the mass conductance varied with $L^{0.76}$ and $G^{0.17}$.

$f_a a = 0.19 L^{0.76}$ (constant air rate and inlet water temperature).. (10)

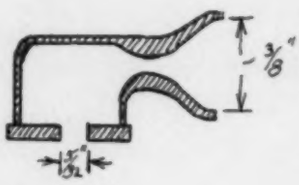
$f_a a = 0.043 L^{0.76} G^{0.17}$ (constant inlet water temperature)..... (11)

If a thermal circuit is drawn it is seen that the conductances from the drops and from the sheets are in parallel and consequently are directly additive, subject to several conditions. First, for a given amount of water down the tower only a certain percentage passes down the walls; this connection may be made approximately by using the water distributions previously obtained. Second, and probably more important, in the case in which the spray is operating, the water from the walls is evaporating into air which contains water droplets and the potentials are therefore not so great as for sheet cooling alone. The greatest amount of mass transfer takes place at the water entrance; consequently on the counterflow side it should be possible to assume that in the region below the level where the spray strikes the walls humidity conditions do not differ too widely from the case in which only sheet cooling occurs. Subtracting magnitudes on the sheet cooling curves (corrected for space water distribution) from magnitudes on the over-all cooling curves will then give the conductance from the spray alone. The results obtained after such procedure indicate that for the sprays the conductance varies linearly with the liquid rate and is relatively unaffected by changes in air rate. This result is confirmed by the work of Hixson and Scott (5) who found that the coefficient for ammonia absorption



1. Nozzle *A*

This nozzle has a machined circular inlet section, a machined mixing chamber, and a machined, conical orifice. The water is forced in on one side of the mixing chamber, thus giving the water a centrifugal motion. It is then forced downward and out through the orifice which is the same size as the inlet. The spray produced is extremely fine, but the spray cone angle is wide.

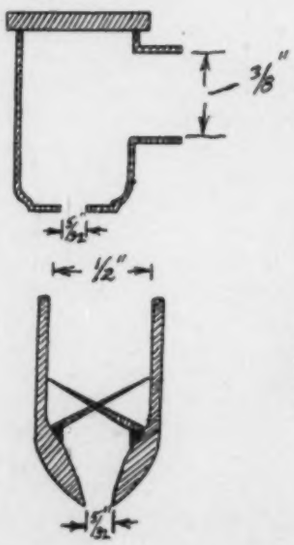


2. Nozzle *B*

This nozzle is the same as the above except that the conical orifice has been plugged with solder and a straight hole drilled through. The spray produced is not quite so fine and is not so evenly distributed about the spray cone. The spray cone angle, however, is reduced about ten degrees.

3. Nozzle *C*

A piece of copper tubing, flared at the end, is soldered to the orifice face of Nozzle A in order to cut down the spray angle. The desired result is obtained, the angle being reduced by about twenty degrees, but again the spray produced is much coarser than that produced by the original nozzle.



4. Nozzle *D*

Nozzle D has a large rough entrance section, a machined finish mixing chamber, and a thin cylindrical outlet orifice. The drops produced are rather large and unevenly distributed about the spray cone.

5. Nozzle *E*

This nozzle is designed to produce a flat spray which is evenly distributed across the cone. The drops produced are rather large, however. The nozzle consists of a straight entrance section, a movable piece forced to rotate by the liquid flow, and a mixing chamber.

Fig. 5—Description of spray nozzles

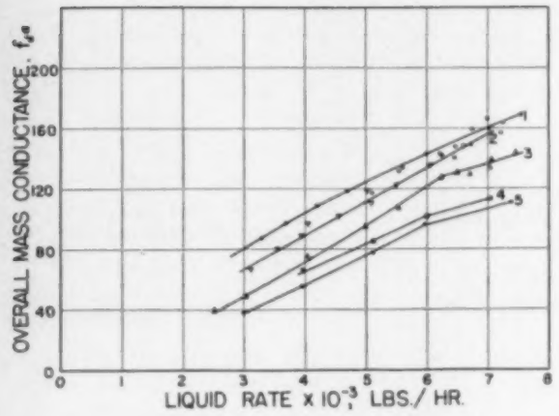


Fig. 6—Overall mass conductance, f_{da}, lb/ft³ hr (lb/lb) versus liquid rate, lb/hr

Air rate = 5000 lb/hr
Legend:
1. Nozzle *A*
2. Nozzle *B* with cylindrical bore
3. Nozzle *C* with 1 in. nipple
4. Nozzle *D*
5. Nozzle *E*

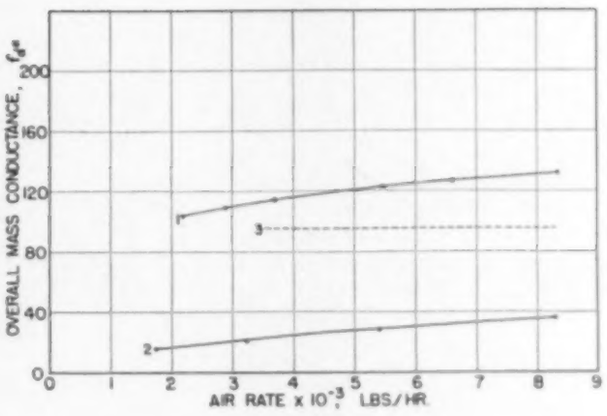


Fig. 8—Overall mass conductance, f_{da}, lb/ft³ hr (lb/lb) versus air rate, lb/hr

Liquid rate equals approximately 5000 lb/hr
Legend:
1. Conductance for spray and walls combined
2. Conductance for walls alone
3. Conductance for sprays alone (obtained by subtraction)

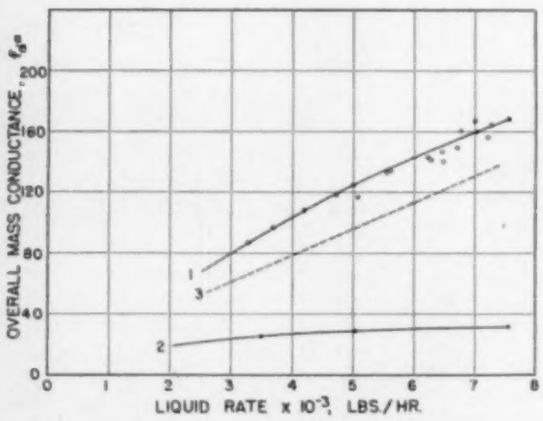


Fig. 7—Overall mass conductance, f_{da}, lb/ft³ hr (lb/lb) versus liquid rate, lb/hr

Air rate equals approximately 5000 lb/hr
Legend:
1. Conductance for spray and walls combined
2. Conductance for walls alone
3. Conductance for sprays alone (obtained by subtraction)

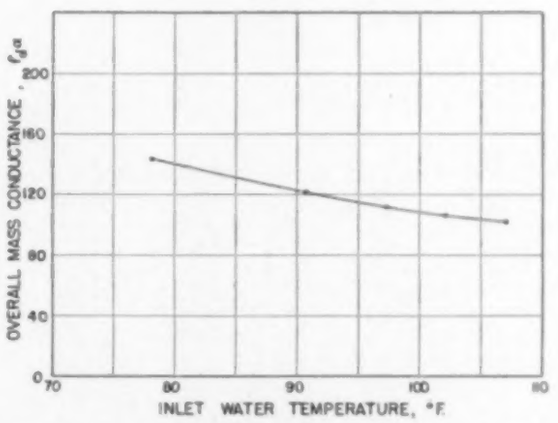


Fig. 9—Overall mass conductance, f_{da}, lb/ft³ hr (lb/lb) versus inlet water temperature, F

Liquid rate equals approximately 5000 lb/hr
Air rate equals approximately 5000 lb/hr

in a spray tower with the wall effect removed varied directly with the liquid rate. The fact that the conductance from the drops is unaffected by changes in the air rate is readily explained in that the relative velocity of the drops and the air stream is dependent mainly on the water velocity (approximately 100 to 150 fps at 5,000 lb per hour) and only slightly on the air velocity (5 ft per second, maximum).

The temperature of the inlet water (for a given tower and air inlet conditions) has been found to have a profound effect upon the over-all mass transfer coefficient. Haslam, Hershey, and Keen (4) in their work on ammonia absorption found that increasing the temperature of the fluid caused the coefficient of absorption to decrease as the 1.4 power of the absolute temperature. Dwyer and Dodge (1) indicate that the coefficient of absorption varies inversely with temperature. In order to determine the effect in the cooling tower, several runs were made holding the liquid and air rates constant and changing only the inlet temperature of the water. The results are indicated in Fig. 9. When plotted on log-log paper this curve yields a slope of -1.09 which compares favorably with the results of Dwyer and Dodge. The equation for the conductance may now be written

$$f_{da} = 5.89 \frac{L^{0.70} G^{0.57}}{T_1^{1.40}} \quad \ldots\ldots\ldots (12)$$

Whenever a liquid and gas come into contact there exists on the gas side of the interface a layer of gas in which motion by convection is slight compared to that in the main body of gas. Similarly on the liquid side there is a surface layer which is practically free from convection. These two layers offer resistance to the flow of heat and material. The resistances are dependent upon the thicknesses of the layers, which in turn are functions of the ratio of viscosity to density (7) (3). Since the viscosity and density are dependent upon the temperature, it was thought that the effect of temperature might be due to its effect on these variables. Kowalke, Hougen, and Watson (6) state that the transfer coefficient for ammonia decreases with increase in temperature

in the same order of magnitude as the viscosity increase of the gas. The same authors found experimentally that the liquid film resistance is approximately one fiftieth of the gas film resistance.

A search of the literature revealed that very little experimental work has been done on the effect of moisture upon the viscosity of air. What little there is, however, (2) indicates that moisture does not change the viscosity appreciably. Gilchrist (2) obtained the following results for dry and saturated air:

Dry air (20.2 C) = 1.811×10^{-7} gms/cm. sec. (Average of four values which agreed within 0.3 per cent).

Sat. air (20.2 C) = 1.816×10^{-7} gms/cm. sec. (Average of three values which agreed within 0.3 per cent).

Calculation of the ratios of viscosities to densities, using viscosity of dry air and density of the air-water vapor mixture shows that the change with temperature does not account for the great change in over-all conductance with temperature.

Another possibility that presents itself is that the liquid side resistance might be appreciable. In the case of water running down the walls of the tower, the assumption of no resistance on the liquid side is probably justified, since the water is thoroughly mixed as it tumbles over the projections placed there for that purpose. The water drops, however, might present another picture. It has been suggested (10), (12) that as the drops fall through the tower they experience little internal mixing, the rate of energy transfer then being dependent upon the transient behavior of the drops. It would be expected, however, that increased temperature would cause convection currents within the drops as they fall and thus increase the internal mixing. The two effects, consequently, are opposed and probably offset each other. In Fig. 10 the operating and equilibrium lines on a temperature-enthalpy diagram (8), (9), (13) are shown for the runs in which inlet water temperature alone was changed. It is readily seen that as the temperature is increased the change in enthalpy of the air passing through the tower is increased. It may also be noted that the operating line moves further away from the equilibrium line, *i.e.*, the driving potential becomes larger.

In the calculation of the conductance, the conductance varies directly as the enthalpy change of the air and inversely as the enthalpy potential.

$$f_d a = \frac{G}{V} \frac{\Delta h}{(h_1 - h)_{lm}} \quad \ldots\ldots (13)$$

If $(h_1\text{-}h)_{lm}$ increases more rapidly than Δh as the temperature is increased, then the conductance will decrease. This, apparently, is what actually occurs.

Lewis Modulus

Various experimenters have found that the ratio of the conductance for heat transfer to the conductance for mass transfer is equal to the specific heat of the air stream.

$$f_c/f_d = s \quad \ldots\ldots (14)$$

If the transfer areas for mass and sensible heat are the same, then

$$f_c a/f_d a = s \quad \ldots\ldots (15)$$

It was found that for the spray tower tested this relationship does not often hold. The failure of the data to yield good checks of the Lewis Modulus (for runs in which the sprays were operating) is probably due to the complete vaporization of small drops. The consequence of this vaporization would be to increase the mass conductance, $f_d a$, and decrease the thermal conductance, $f_c a$, thus yielding low values of the Lewis Modulus, as found experimentally. It was also found that in the cases in which the Lewis Modulus did hold, the change in temperature of the air passing through the tower was comparatively large. This merely means that the effect of small drop vaporization (on the thermal conductance) is negligible when the thermal conductance is large.

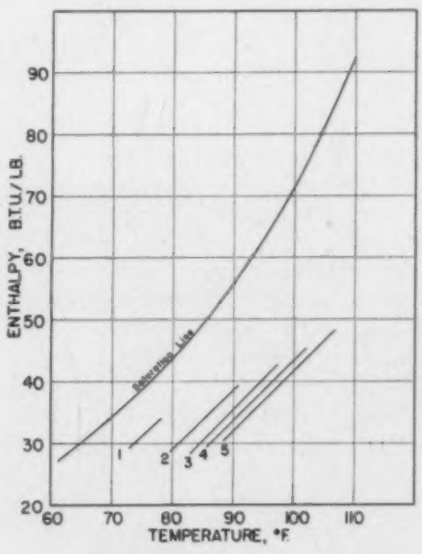


Fig. 10—Effect of inlet water temperature on the operating line. Enthalpy, Btu/lb versus temperature, F

Legend:
1. Run No. 66
2. Run No. 38
3. Run No. 63
4. Run No. 64
5. Run No. 65

Effectiveness

The effectiveness of the tower, calculated on an enthalpy basis (8) is plotted as a function of the ratio of liquid rate to air rate in Fig. 11.

$$\epsilon = \frac{h_1 - h_2}{h_{11} - h_2} \quad \ldots\ldots (16)$$

The equation of the average line drawn through the experimental points is

$$\epsilon = 0.125 + 0.27 \times \frac{L}{G} \quad \ldots\ldots (17)$$

The plotted magnitudes have not been corrected for change in inlet water temperature which varied from 90 F to 100 F.

Conclusions

1. In a spray tower the performance is affected more by the drop size distribution than by the water distribution within the tower. The greatest amount of cooling is obtained in the section where the water first enters.

2. The conductances '$f_d a$' determined for this spray tower should not be extrapolated too widely to other tower volumes, for the drop size distribution and the liquid distribution in the tower are important variables which have not yet been included in the correlation.

3. The conductance for mass transfer from the walls of the tower varies as $G^{0.5}$ and $L^{0.2}$.

4. The over-all conductance for mass transfer from both spray and walls (*i.e.*, over the whole tower) may be represented by the empirical equation

$$f_d a = 5.89 \frac{L^{0.78} G^{0.17}}{T_1^{1.60}} \quad \ldots\ldots (18)$$

5. The mass conductance for the spray alone varies linearly with the liquid rate and is unaffected by the air rate.

6. Complete vaporization of small drops causes the ratio of $f_c a/f_d a$ to be smaller than the humid heat of the air stream.

Acknowledgment

The writers desire to express their appreciation to H. Eagles and S. Hori for their aid in the performance of the laboratory work, to the Golden Gate Chapter of the AMERI-

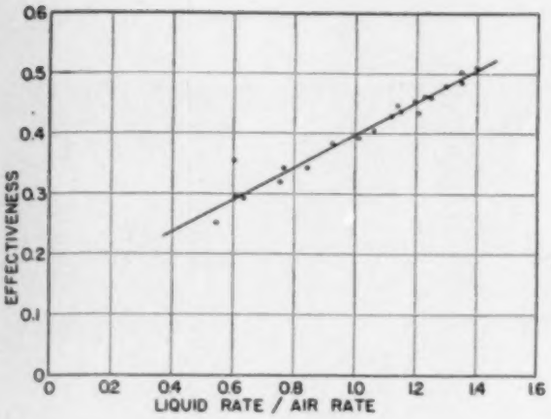


Fig. 11—Effectiveness, $\frac{h_1 - h_2}{h_{i1} - h_2}$ versus ratio of liquid rate to air rate

Inlet water temperature = 90 to 100 F

CAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and the San Francisco Air Conditioning Society for their interest in the work. The W. R. Ames Co., San Francisco, made the cooling tower available to the University of California.

Nomenclature

a = transfer area per unit volume of tower, ft²/ft³

c = unit heat capacity of liquid water, Btu/lb F

f_c = over-all unit thermal conductance for convection, Btu/ft² hr F

f_d = over-all unit conductance for diffusion, lb/ft² hr (lb/lb)

f_h = over-all unit conductance for enthalpy transfer, Btu/ft² hr (Btu/lb)

G = dry air rate, lb/hr

h = air-water vapor-mixture enthalpy, Btu/lb

Δh = change in enthalpy of air passing through tower, Btu/lb

H = air humidity, lb of water vapor/lb of bone dry air

ΔH = change in humidity of air passing through tower, lb/lb

L = liquid water flow rate, lb/hr

ΔL = change in basin level, ft

s = humid heat of air-water vapor mixture, Btu/F lb of bone dry air

t = air main body temperature, F

Δt = change in temperature of air passing through tower, F

t_{WB} = air main body wet bulb temperature, F

T = water main body temperature, F

V = tower volume = 35 ft³

NTU = number of transfer units = $\frac{f_d a V}{G}$

Subscripts

1 = tower conditions at water entrance and air exit

2 = tower conditions at water exit and air entrance

i = interface conditions

lm = logarithmic mean

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ENGDAHL JOINS BATTELLE

Richard B. Engdahl, Urbana, Ill., has joined the research staff of Battelle Memorial Institute, Columbus, O., and has been assigned to the division of fuels research.

Mr. Engdahl was special research assistant and member of the faculty at the University of Illinois. He is a member of the ASHVE.

ASHVE Officers and Council Members meet Mayor of San Francisco at 1st Pacific Heating and Air Conditioning Exposition. (*Left to right*): W. A. Russell, E. K. Campbell, A. J. Offner, W. L. Fleisher, Mayor Angelo Rossi, E. O. Eastwood, M. F. Blankin, L. P. Saunders and F. E. Giesecke

Candid Camera Shots

Made at the

First Pacific Heating and Air Conditioning Exposition

An Analysis of Factors Influencing Building Heat Losses

By Paul D. Close*, Chicago, Ill.

ONE problem that is of greatest importance to heating engineers is the calculation of heat losses, but it is probable that no single heating problem involves as many factors and variables which are so difficult to evaluate.

Heat losses calculated by various engineers for the same building will show a wide variation in results. In one instance with which the writer is familiar, five different sources, including a heating contractor and a gas heating company, calculated the heat losses of a residence and five widely different results were obtained, the maximum result being almost twice the minimum. In another instance, the calculated heat losses for a residence obtained by a public utility were more than 50 per cent greater than the results obtained by a qualified heating engineer using THE GUIDE method. While it is expected that there would be some variation in the results due to different assumptions and to discrepancies in estimating areas and other quantities, it would seem that there should be reasonable agreement between the several methods now in common use.

The purpose of this paper is to analyze some of the factors and variables which must be evaluated, to discuss some of the differences in the methods used for calculating heat losses and to offer certain recommendations. In considering this subject, it might be well first to review the fundamentals involved. The unknown quantities to be determined may be either (1) the *maximum* hourly heat loss for each room or space to be heated based on maintaining a specified inside temperature, or (2) where the problem is one of seasonal fuel consumption, the *average* hourly heat loss for the entire building.

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For a building in operation the hourly Btu requirement includes (1) *the transmission losses* or heat losses due to the transmission of heat directly through the building materials, and (2) *the infiltration losses* or the heat losses due to air leakage through cracks, crevices and openings. The infiltration losses may include any heat required to vaporize the moisture added for humidification requirements. For intermittently heated buildings, another quantity must be considered, namely, the amount of heat required to raise the temperature of the building and its contents to the specified inside temperature. Heat loss calculations are based on constant inside and outside conditions, although actually the outside conditions are varying continually.

Transmission Losses

The heat loss by transmission through any surface for equilibrium conditions is proportional to the coefficients of transmission U, to the net area A, and to the inside-outside temperature difference $(t\text{-}t_o)$. To estimate the transmission losses it is only necessary to know the net areas of the surfaces involved, the coefficient of transmission of these surfaces and the inside and outside temperatures. The surface area may, of course, be obtained from the plans or from measurements of the actual building. Reliable heat transmission coefficients for constant or equilibrium conditions are given in THE ASHVE GUIDE. While these are calculated coefficients, check tests have established the fact that they are reasonably accurate or within the allowable accuracy range. The inside temperature is, of course, known and the outside temperature must be selected for the locality involved, based on weather bureau records.

Wind Velocity and Surface Coefficients

Wind velocity does not enter directly into the calculation of the transmission losses and influences the transmission losses only to the extent that it affects the outside surface conductance. Fig. 1 shows the relation between wind velocity and surface resistance (the reciprocal of the surface conductance) for average building materials. The surface resistance decreases from about 0.61 for still air to 0.17 for a 15 mph wind velocity and to about 0.10 for a 30 mph wind velocity. In the case of constructions of low heat transmission value or high resistance, the effect of wind velocity is a negligible factor because the ratio of the surface resistance to the total or over-all resistance is small. As the over-all heat loss coefficient increases, the relative effect of the velocity on the over-all heat loss increases, because the surface resist-

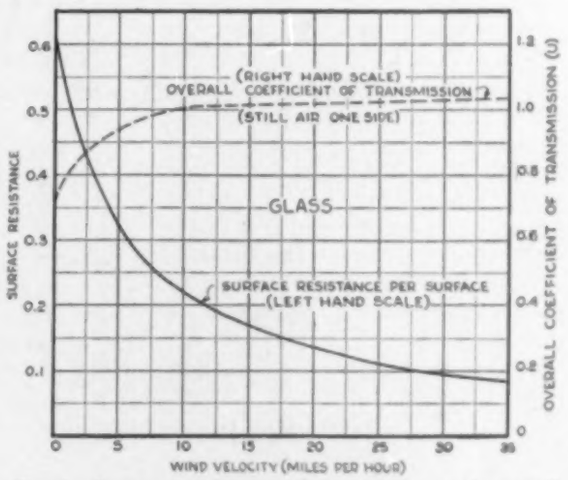

Fig. 1—Curves showing surface resistances and over-all coefficients of transmission for single pane glass, for various wind velocities

ance comprises a greater proportion of the total resistance. Thus the wind velocity may constitute a relatively important factor in the case of the transmission through thin, highly conductive materials such as window glass, but has a minor effect on constructions of average heat loss value.

The over-all wall coefficients given in THE ASHVE GUIDE are based on a wind velocity of 15 mph, the surface conductance for this wind velocity being 6.00 and the resistance 0.17. For higher wind velocities, there is relatively little decrease in the surface resistance as will be apparent from Fig. 1. For wind velocities less than 15 mph the actual surface resistance will be higher than that for 15 mph, so that the theoretical wall coefficient used for the calculations will be lower than it should be. The difference is small, however, and is on the side of safety. For example, a wall having a coefficient of 0.20 for still air will have a coefficient of 0.219 for a 15 mph wind velocity and 0.222 for a 30 mph wind velocity, but in most localities a 15 mph wind velocity is nearer the average than still air (0-mph wind velocity).

Infiltration Losses

The heat loss by infiltration is a function of the volume of outside air entering the building in cubic foot per hour Q, the density and specific heat of the air and the inside-outside temperature difference $(t\text{-}t_o)$. All of the factors are known or can be determined with reasonable accuracy except Q, the volume of outside air entering the building per hour. This quantity depends on the wind velocity and direction, the width of crack or size of openings, the type of openings and other factors. Since the air entering at any given point must displace an equal volume of air leaving at some other point, the number and size of the openings on the leeward side of the building or elsewhere may have as much effect on the infiltration as the cracks or openings on the windward side.

Back Pressure

To allow for the building up of pressure within rooms due to the resistance against which the infiltration of air must operate, it is customary to reduce test values through cracks or openings of various sizes by a certain percentage, usually 20 per cent. This is a purely arbitrary figure for it is obvious that the resistance against which the infiltration of air must operate depends on many factors and conditions and, therefore, will vary over a considerable range. It is conceivable that the amount of infiltration might be practically nil, even through fairly large cracks or openings, if the building were so tight elsewhere as to preclude the exfiltration of the displaced air.

Amount of Window Crack

Another factor which complicates the problem of estimating air leakage is the amount of crack to be selected. If a room has only one window, whatever air enters through the cracks of this window cannot leave through the same cracks but must find egress elsewhere in the room or building. The extent to which this may take place depends on whether the door or doors to the room are open or closed, and if closed, the degree of tightness. The present ASHVE practice is to use all the crack for a room with windows on one side. For a room with windows on two sides the wall having the most crack is selected whereas with three or four sides exposed the practice is to take the wall having the most crack but not less than ½ the total crack. These are purely arbitrary rules based on engineering judgment and experience.

Conditions in Practice vs. Test Data

Still another difficulty is to know whether the conditions such as crack width assumed for the building for which the estimate is being made agree with the conditions on which the test data being applied, were based. For example, the air leakage for an average non-weather-stripped unlocked double-hung wood sash window with $\frac{1}{16}$ in. crack and $\frac{3}{64}$ in. clearance, is given in THE GUIDE 1941 as 39.3 cu ft per foot of crack per hour for a 15 mph wind velocity. This includes the wood frame leakage. The question is, how nearly do these conditions conform to the actual conditions of the building involved? Suppose the actual width of crack were 50 per cent greater or $\frac{3}{32}$ in. and the clearance were twice as much, or $\frac{3}{32}$ in. This would be regarded as a poorly fitted window, much poorer than average. In this case, the air leakage according to the GUIDE data would be 110.5 cu ft per foot of crack per hour, or nearly three times as much as for $\frac{1}{16}$ in. crack. It is apparent, therefore, that much depends on the degree with which the actual conditions in practice coincide with the conditions assumed. The results may be considerably in error if there is a serious discrepancy between the actual conditions and the test conditions.

Infiltration Allowances

As previously stated, the air entering one side of a room may leave through windows on the other side of the room. It is also possible that air leakage may take place through windows on two sides of a room and the displaced air leave through cracks in other rooms on the leeward side. To provide adequate heat in each room or space, an allowance must be made for air leakage for each such room or space having windows and doors, although there may be some question as to where the displaced air may leave at any given moment. Since, however, air leakage does not take place from all rooms or spaces at the same time, the sum of the infiltration allowances for a building may be approximately twice as high as will actually take place for the entire building, if the allowance for each room or space is adequate when the room or space is on the windward side.

Infiltration Most Difficult Variable to Estimate

The heat required to warm the infiltering air to the specified inside temperature can be calculated with accuracy. But it is apparent that the most difficult problem is to estimate the amount of air leakage. This is probably the major source of error in the problem of calculating heat losses.

Practical Application of Data

Even though it were possible to estimate the amount of air leakage with accuracy and all other constants and coefficients were known, there is still another important phase of the problem, namely, the

Table 1—Infiltration Losses
(Btu per hour)

Temp. Dif.	Wind Velocity (Mph)						
	0	5	10	15	20	25	30
Q[a]	0	125.5	406	747	1126	1250	1935
20	0	46	146	269	405	548	696
40	0	91	292	538	810	1096	1392
60	0	137	438	806	1215	1644	2088
80	0	182	584	1075	1620	2192	2734
100	0	228	730	1344	2025	2740	3480

[a] Air leakage, cubic feet per hour

practical application of the data. The outside temperature and wind velocity do not affect the heat losses in the same ratio or degree. Both the transmission and infiltration losses are proportional to the temperature difference. The infiltration losses are also proportional to the wind velocity whereas the transmission losses, as previously stated, are only influenced to a minor extent by wind velocity except in the case of thin materials having a high coefficient of transmission. If a building had no windows, there would be little or no air leakage and practically the total heat loss would be that due to transmission. On the other hand, if a building had a large percentage of windows, the infiltration loss would be correspondingly high and might be several times the transmission loss.

The problem in any case is to select that combination of temperature and wind velocity which will result in the maximum heat loss for the building in question. Because of differences in the ratios of the transmission and infiltration losses, any given combination of outside temperature and wind velocity, such as a —10 F temperature and a 15 mph wind velocity will not affect the heat losses of different buildings alike. This combination may result in the maximum heat loss for a certain building, but another combination may result in the maximum heat loss for other buildings. For a building with no windows, the maximum heat loss will occur during the coldest days, but this is not necessarily so for a building with an average or high percentage of windows — the maximum heat loss in this case may occur on a windy day when the temperature is considerably higher, since the infiltration losses may in the case of such a building result in the maximum heat loss.

An example may serve to illustrate the different effects of temperature and wind velocity on the transmission and infiltration losses. Fig. 2 is a plan of a room 15 x 20 ft having two double-hung unlocked non-weatherstripped windows with an assumed crack of $\frac{1}{16}$ in. and $\frac{3}{64}$ in. clearance. The walls are assumed to have an over-all coefficient *based on still air* of 0.20. The infiltration heat losses for various wind velocities and temperature differences are given in Table 1, the air leakage

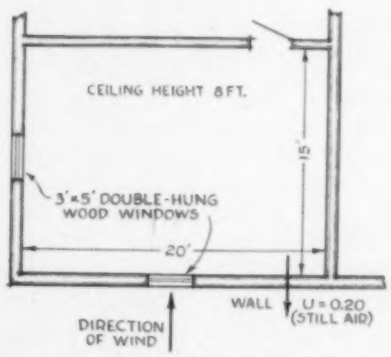

Fig. 2—Plan of 15 x 20 ft room with two 3 x 5 ft double-hung wood windows

value Q being based on values given in The ASHVE Guide 1941, Chapter 4. The wall transmission losses are given in Table 2 and the glass transmission losses in Table 3. The wall and glass coefficients U for various wind velocities are also indicated in this table.

These infiltration and transmission losses are plotted in Fig. 3. The lower curves show the infiltration losses to be proportional to the wind velocity as expected. The upper curves showing the transmission losses indicate the comparatively minor effect of wind velocity on the wall losses but somewhat greater effect on the glass transmission losses.

Fig. 4 shows the combined transmission and infiltration losses for various outside temperatures and wind velocities. It is interesting to note that, for this room and the conditions assumed, the calculated heat loss of 5750 Btu for an outside temperature of —10 F and for still air (no wind) is approximately the same heat loss as for an outside temperature of +25 F and a wind velocity of 30 mph, a difference of 35 deg in the outside temperature for the same heat loss, due to the increase in wind velocity.

Selecting Temperatures and Wind Velocities

The curves of Fig. 4 could be used for estimating the maximum heat loss for this room for any locality by selecting the proper temperature and wind velocity for that locality. According to the ASHVE method, the temperature selected is a certain number of degrees above the lowest on record and the wind velocity is the average during the three coldest months, or December, January and February. In Chicago, for example, the design temperature usually is —10 F and the wind velocity generally is 12.5 mph, the average during the three coldest months. For these conditions, the heat loss for this room according to Fig. 4 is about 7800 Btu per hour. The question is, does this quantity represent the most probable maximum heat loss for this room? This loss is based on an *average* wind velocity. Would it not be equally as

Table 2—Transmission Losses (Walls)[a]
(Btu per hour)

Temp. Dif.	Wind Velocity (Mph)						
	0	5	10	15	20	25	30
U	0.20	0.21	0.22 (0.217)	0.22 (0.219)	0.22	0.22 (0.222)	0.22 (0.223)
20	1000	1050	1085	1095	1100	1110	1115
40	2000	2100	2170	2190	2200	2220	2230
60	3000	3150	3255	3285	3300	3330	3345
80	4000	4200	4340	4380	4400	4440	4460
100	5000	5250	5425	5475	5500	5550	5575

[a] Net area, 250 sq ft.

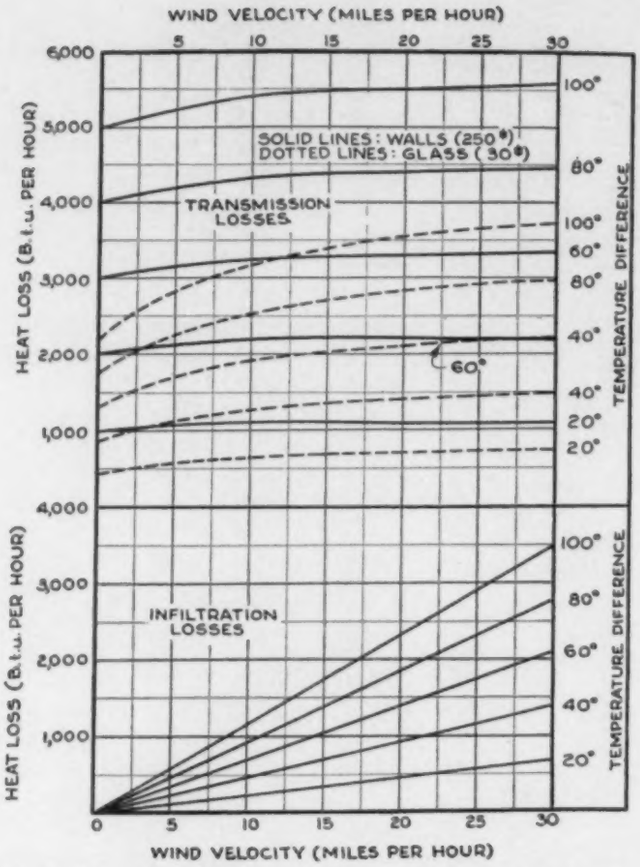

Fig. 3—Curves showing infiltration losses and transmission losses for room shown in Fig. 2, for various wind velocities
(The solid lines of the upper curves show the wall transmission losses and the dotted lines show the glass transmission losses.)

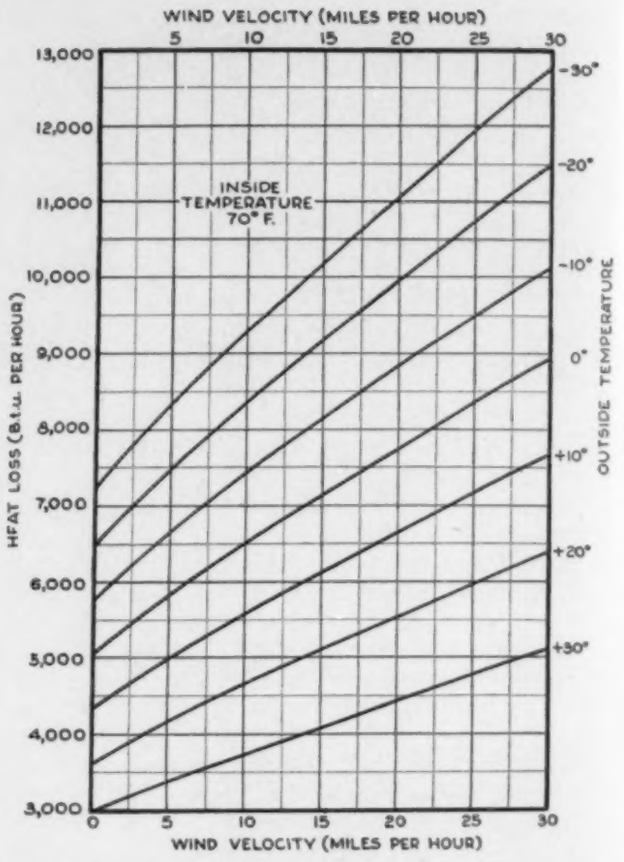

Fig. 4—Total heat losses for room shown in Fig. 2 for various wind velocities and outside temperature (inside temperature, 70 F)

logical to select a near maximum wind velocity and an average temperature? The problem in any case is to ascertain that combination of temperature and wind velocity which, operating together, will produce the maximum heat loss.

The ideal or theoretical solution would be to calculate the heat losses for a building on the basis of various combinations of temperature and wind velocity which have occurred in the past, based on weather bureau records, and the combination which resulted in the maximum heat loss would be the heat loss for the building in question. Even such a procedure would not preclude the possibility of a combination of weather conditions developing which had never before been experienced.

Until such time as the procedure of using concurring combinations of temperatures and wind velocities can be simplified it will probably be necessary to calculate heat losses on the basis of a single assumed combination of values for each locality. But whether the assumed outside temperature should be a certain number of degrees above the lowest on record and the wind velocity the average for the three coldest months of the year, in accordance with present ASHVE recommended practice, is not entirely certain. At any rate, because of the variable effect of temperature and wind velocity on the heat losses of various types of buildings, the proper selection of temperatures and wind velocities in various localities and for different buildings may be a difficult problem.

Exposure Factors: Direction of Wind

Some organizations recommend the application of an exposure allowance for rooms exposed to prevailing winds. At one time THE GUIDE recommended the use of a 15 per cent exposure factor. However, experience seemed to indicate that the calculated heat losses were more than adequate in most cases without the use of any additional factors for rooms subjected to prevailing winds.

While rooms on the windward side will require more heat than those on the leeward side, other things being equal, it is the writer's opinion that the same provision should be made for the wind blowing on all sides of a building, because a sudden change in the wind direction will reverse the windward and leeward rooms, and the room which one hour was on the leeward side may in the next hour require the same heat demand as a room exposed to prevailing winds. If each room is calculated for the wind and temperature conditions to which it will be subjected when the wind is blowing against that room, then it would seem that no additional allowance need be made. The primary effect of wind velocity, as previously indicated, is on the infiltration losses, and each room or space should, if properly considered, be provided with sufficient heat to take care of the estimated air leakage for that room independently of all others.

Table 3—Transmission Losses (Glass)[a]
(Btu per hour)

Temp. Dif.	Wind Velocity (Mph)						
	0	5	10	15	20	25	30
U	0.73	0.94	1.05	1.13	1.18	1.22	1.24
20	438	564	630	678	708	732	744
40	876	1128	1260	1356	1416	1464	1488
60	1314	1692	1890	2034	2124	2196	2232
80	1752	2256	2520	2712	2832	2928	2976
100	2190	2820	3150	3390	3540	3660	3720

[a]Net area, 30 sq ft.

Relation of Temperature to Wind Velocity

A rule was developed by one organization in the heating field to the effect that one mile of wind velocity was equivalent to one degree temperature change. While it is conceivable that such a relationship might be approximately correct in certain instances, it seems probable that this relationship would more often be incorrect. Referring again to Fig. 4, an outside temperature of —20 F and still air is equivalent to a temperature of +20 F and a wind velocity of about 31 mph. In this instance an increase of 31 mph in the wind velocity is equivalent to a temperature change of 40 F, or one degree temperature change is equivalent to ¾ mph wind velocity. However, if this room had no windows, there would be comparatively little effect due to the change in wind velocity as evidenced by the upper solid curves of Fig. 3, showing the wall transmission losses. For example, a 30 mph change in wind velocity is equivalent to a temperature difference increase of from about 80 to 89 F, or 9 deg temperature change is equivalent to about a 30 mph wind velocity change, a ratio of less than one to three. The ratio is even smaller for lower temperature differences.

Minimum Wall Coefficients

At least two associations are now recommending a minimum wall coefficient of 0.10 for outside frame walls. In one instance, this was due to the fact that a check on field results indicated higher heat loss coefficients than the minimum theoretical calculated values. The use of a minimum coefficient of 0.10 will provide a factor of safety against defects in construction or in the application or functioning of the insulating material. The lower the theoretical wall coefficient, the greater will be the percentage of error resulting from defects in construction or failure of the insulating material to perform as rated.

Solar Radiation

There are many incidental factors governing the heat losses from buildings. Solar radiation, for example, has a definite influence on daily or seasonal fuel consumption but should not be taken into consideration in arriving at the maximum heat demand for it must be assumed that this maximum demand will occur at night.

Heat Capacity and Dampness

The heat capacity of the building materials may also have a bearing on the heat requirements although this item is not usually considered except in the case of intermittently heated buildings.

The moisture content of the building materials may be increased by precipitation and thereby increase the rate of heat flow through them. This factor, however, should not otherwise appreciably influence the heat losses.

Basement Floor and Wall Heat Losses

In many cases no heat is provided in the basement and consequently the heat losses are not calculated for this space. However, where a specified inside temperature is to be maintained as in the case of recreation rooms, the heat losses through the basement floor and walls must be calculated. The heat loss coefficients for basement floors have been based on the assumption that the coefficient includes only the resistance of the materials above the ground whereas it has been apparent that the heat resistance of the ground should be taken into consideration. Where the basement heat losses have been calculated this has resulted in a substantial error.

A program now in progress at the ASHVE Research Laboratory in Pittsburgh, with the cooperation of the Insulation Board Institute, is expected to provide more accurate data on basement floor and wall heat losses. Results to date indicate that the calculated heat losses for uninsulated basement floors are ten times or more greater than the actual heat losses obtained by these tests. The ratio of this discrepancy, however, apparently diminishes for basement walls as the top of the wall is approached. Both the heat capacity and the conductivity (that is, the diffusivity) of the dirt in contact with basement floors and walls apparently govern the heat losses through such floors and walls.

Tall Buildings

Stack effect must be considered in the case of tall buildings. Higher wind velocities also prevail at high levels.

Attic Ventilation

Attic ventilation is being recommended as a precaution against condensation in attics. Such ventilation will partially nullify the heat resistance value of the roof and thereby lower the attic temperature, which in turn increases the ceiling heat loss. If the ceiling has a low heat transmission coefficient, the elimination of the roof will have only a minor effect upon the over-all heat loss through the ceiling and roof, but as the ceiling coefficient increases, the relative value of the roof

Table 4—Total Heat Loss
(Btu per hour)

Temp. Dif.	Wind Velocity (Mph)						
	0	5	10	15	20	25	30
20	1438	1660	1861	2042	2213	2390	2555
40	2876	3319	3722	4084	4426	4780	5110
60	4314	4979	5583	6125	6639	7170	7665
80	5752	6638	7444	8167	8852	9560	10220
100	7190	8298	9305	10209	11065	11950	12775

increases. The combined ceiling and roof coefficient will without ventilation often be equivalent to a substantially lower ceiling coefficient with ventilation. Where the roof and ceiling coefficients are of average value, it is more effective from the winter loss standpoint to avoid attic ventilation, thereby retaining the insulating value of the roof. With the heat resistance more evenly divided between the ceiling and roof, the problem of attic condensation is not so serious, and in any event can be guarded against by installing a vapor barrier in the top floor ceiling. Attic ventilation is particularly desirable in the summer but may not always be necessary in the winter, especially if vapor barriers have been installed in the ceiling. Therefore, in the interest of economy, it is recommended that such ventilation be reduced in the winter where possible, except where it is necessary to preclude attic condensation. In any event, attic ventilation may seriously influence the heat loss through the top floor ceiling and should be given proper consideration.

Summary, Conclusions and Recommendations

1. The problem of calculating heat losses does not always lend itself to an exact or precise solution. By the very nature of the factors which must be considered, the result can only be regarded as an approximation of the probable heat losses for the conditions assumed.

2. Perhaps the most difficult quantity to be evaluated is the amount of air leakage. Although it is quite possible to predetermine the transmission losses with reasonable accuracy, the infiltration losses may be considerably in error and undoubtedly constitute the major source of error.

3. Another reason for possible inaccuracies is the difficulty of applying the transmission and infiltration factors so that the proper combined effect of temperature, wind velocity and other variables is obtained. The outside temperature and wind velocity used in the calculations should theoretically be that weather bureau combination which together will produce the maximum combined transmission and infiltration losses.

4. There are many incidental factors which may also influence the heat losses of a building, most of which are generally neglected in the calculations.

5. Various short-cuts involving the use of charts or tables have been devised, but these generally sacrifice accuracy and are even more approximate than the more studied methods. Practically all methods, including so-called rules-of-thumb *work* because the results usually are on the safe side and also because in most instances, there is a factor of safety in the excess capacity of the heating equipment installed. While it is desirable that the methods be simplified as much as possible, such simplification should not sacrifice accuracy, for at best the results are none too accurate.

6. The average heat demand during the heating season usually is less than half the maximum for which the heating system is designed and it is only during the comparatively short periods of extreme weather conditions that the accuracy of the calculated heat losses and the adequacy of the installed heating equipment may be tested. Even then the accuracy of the calculated results cannot be judged unless the equipment is inadequate. As previously stated, the results in most cases are well on the side of safety.

7. There are several methods in use for calculating heat losses including those of the ASHVE, *HPACCNA, NWAHACA, IBRM* and the *AGA*. Practically all methods employ the use of the ASHVE or similar heat transmission coefficients, some with certain modifications and all make certain allowances for air leakage. But there are often considerable differences in the results obtained by these methods due to differences in the application of the fundamental factors and in allowances made for various conditions.

8. For any given building and location, there should be only one *most accurate* method of calculating heat losses and it seems logical that all technical societies and associations concerned with this problem should be using substantially the same methods and that the results should be reasonably in agreement.

It is hoped that this paper will stimulate an interest in this subject to aid in the development of a uniform and accurate method of calculating heat losses which will be acceptable to all organizations concerned with this problem.

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Lighting and Air Conditioning Design Factors*

DURING the fall of 1939 the joint committee on Lighting in Air Conditioning, of the *Illuminating Engineering Society* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, initiated a design survey of representative air-conditioning installations. The purpose of this investigation was to bring to the attention of air-conditioning engineers and illuminating engineers the relative magnitude of the various heat gains, general practice in the circulation of air and location of air grilles as related to position of lighting, and the range of design temperatures.

*Report by *IES*-ASHVE Joint Committee on Lighting in Air Conditioning: H. M. Sharp, *Chairman;* E. E. Ashley, A. A. Brainerd, A. D. Cameron, W. P. Friend, W. E. Stark, Walter Sturrock.

Through the cooperation of the *Air Conditioning Manufacturers Association* 82 installations were made available for study, representing a wide geographical selection. The essential data are summarized in Table 1, and from these the interested investigator may obtain what would be of value in any particular study.

The committee realizes that the interest in the material presented in Table 1 will depend upon the major interest of those referring to it. However, the increased interest in the cooling load component of a lighting system, particularly on the part of illuminating engineers working in conjunction with air conditioning designers and architects, would seem to warrant further elaboration and interpretation of Items 8 to 17, inclusive, in that table.

The sensory prominence of lighting has attracted to itself certain unfavorable attention as a source of heat gain and as a factor in discomfort due to sensations of warmth that is not supported by available data.[1,2,3] When confronted with situations of this nature the tendency has been to reduce the lighting component to the lowest possible value, thus disregarding the importance of adequate illumination in

[1]Effects of Artificial Lighting On Air Conditioning, by Walter Sturrock. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 213.)

[2]Cooler Foot Candles for Air Conditioning, by W. G. Darley. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August, 1940.

[3]Mean Radiant Temperatures of Lighting Systems with Comfort Correlations, by Howard M. Sharp (*Illuminating Engineering*, Vol. 35, November, 1940, No. 9.)

Table 1—Lighting and Air Conditioning Design Factors

1. Classification	Office			Restaurants		Apparel Stores				Department Stores		Drug Stores			Beauty Shops		Small Stores		Hotels	Apt. Bldg.
2. Location (section of U.S.A.)[a]	S.W.	S.	N.E.& C.	S.W.	N.E.& C.	S.W.	N.E.& C.	Cal.	Wash.	S.W.	N.E.& C.	S.W.	N.E.& C.	Cal.	S.	N.E.& C.	S.	N.E.& C.	S.	N.E & C.
3. No. of installations reported	3	9	19	1	8	5	10	2	1	2	9	1	1	1	1	1	1	5	1	1
4. Total floor area represented (thousands sq ft)	96.52	457.2	153.0	0.600	22.72	40.35	113.07	46.0	17.30	37.70	353.1	4.45	3.09	6.66	0.425	1.09	3.16	12.02	22.2	4.05
TOTAL COOLING LOAD																				
5. Computed, in tons (average per installation)	124.2	182.1	88.8	7.25	24.59	19.9	38.1	45.2	12.34	58.1	132.0	26.3	5.5	19.3	4.51	5.5	15.2	9.26	143	26.0
6. Computed, tons per 1000 sq ft floor area	4.12	3.58	38.2	12.00	10.2	2.38	3.35	2.17	7.00	3.28	3.32	5.9	1.78	2.9	10.6	1.09	4.80	3.99	6.44	6.32
7. Tons of refrigeration capacity provided	128.6	187.7	65.73	7.00	26.3	21.34	39.9	51.0	12.50	62.7	136	28.0	—	21.3	5.0	—	17.0	10.00	120	26.0
8. Ratio refrigeration installed to computed, in percent	102.5	102.2	112.0	100	105	107.0	104.0	112.0	100	107.0	103	107.0	—	105	117	—	111.0	108	84.0	100
LIGHTING																				
9. Watts in use per sq ft	2.72	2.11	1.876	1.0	2.03	1.75	3.33	3.0	8.1	2.13	3.01	1.8	0.7	1.8	1.06	2.7	5.73	2.28	2.25	3.0
10. Watts in use per cubic foot of conditioned space	0.214	0.20	0.128	0.063	0.162	0.162	0.400	0.297	3.15	0.1613	0.305	0.121	0.05	0.106	0.08	0.22	0.455	0.172	0.225	0.09
11. Ratio of watts in use to watts installed	100	76.4	58.95	100	81.3	85.9	71.7	100	—	100	95.7	80	100	80	100	100	100	100	80	55.9
COOLING LOAD COMPONENTS IN PER CENT OF TOTAL COMPUTED COOLING LOAD																				
12. Lighting	21.5	30.2	14.3	2.3	9.9	21.3	28.0	23.3	47.0	19.6	25.1	8.8	10.9	17.6	2.8	13.6	33.5	17.1	20.0	13.4
13. Solar radiation through windows	9.6	14.9	15.8	4.2	2.0	3.0	5.5	10.9	2.2	8.7	5.9	—	—	—	4.85	—	—	1.6	—	17.30
14. Conducted heat	27.8	19.3	25.8	10.0	13.6	34.5	24.1	29.5	17.4	17.7	15.3	13.0	36.4	18.9	30.6	18.2	14.0	32.5	32.9	50.0
15. Fresh Air	24.8	30.8	22.4	47.6	29.5	23.1	18.7	24.9	12.5	32.8	27.2	35.4	18.2	17.9	14.4	13.6	27.6	25.5	35.7	5.13
16. Occupants	9.8	13.3	12.2	25.0	26.0	14.7	19.7	17.2	18.2	13.8	22.5	33.1	27.3	36.2	15.5	13.6	11.0	15.7	11.5	5.1
17. Miscellaneous Heat Gains	3.0	9.2	9.0	10.0	30.3	3.1	4.5	—	*2.6	6.8	5.0	9.9	18.2	9.4	30.8	36.4	6.8	8.3	—	7.7
AIR DISTRIBUTION																				
18. Fresh outside air in percent of total circulated	14.9	32.72	18.98	51.0	52.3	27.5	15.86	25.4	22.0	18.9	17.8	44.0	100	26.3	14.4	100	11.80	27.1	21.4	100
19. Total air circulated, in cfm per 1000 sq ft floor area	1,941	1,656	1,650	3,660	2,290	1,150	1,440	1,280	3,900	1,210	1,210	1,530	136	2,850	3,430	2,750	2,160	1,800	2,730	148
20. Total air circulated, in cfm per ton of total cooling load	476	424.4	452.3	304	251	477.2	433.2	556	552	367.5	410	250	76.3	984	323	54.5	403	416	432	23
21. Location of Air Grilles																				
Ceiling	.....	.....	6	.....	3	.....	2	3	.....	.....	1	.....	.....	1	1	.....	.....	.....	.....	.....
Sidewalls	3	9	8	1	3	5	7	3	1	2	8	1	.....	.....	.....	.....	1	5	1	.....
Distance above floor in feet	10	9	9	12	.....	10	11	—	9.5	13	12	12.5	.....	.....	13	.....	12	8.5	12	.....
22. Direction of Air Flow from grilles																				
Horizontal	2	9	12	1	3	5	5	3	1	2	7	.....	.....	.....	.....	.....	.....	2	1	.....
Downward	.....	.....	.....	.....	3	.....	3	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Upward	1	.....	.....	.....	.....	.....	.....	.....	.....	.....	1	.....	.....	.....	.....	.....	.....	2	.....	.....
23. Design Temperatures																				
Indoor Dry-Bulb	81	80	80	81.0	80	80	80	79	72	80	80	81.0	80.0	80.0	80	80	78	80	80	79
Indoor Wet-Bulb	66	67	67	65.9	67	64	67	66	60	67	67	65.9	66.0	66.5	66	66.0	66	67	67	66
Outdoor Dry-Bulb	101	81	95	96.0	95	98	95	93	83	100	95	96.0	95	95.0	80	95	95	95	100	95
Outdoor Wet-Bulb	77	77	77	76.0	75	76	76	71	61	78	78	76.6	75	70.0	65	75	79	76	78	77

[a] Abbreviations refer to sections of country as follows: N.E. & C.—Pennsylvania, Michigan, Illinois, West Virginia and New York; S.—Louisiana, Georgia, Tennessee and Kentucky; S.W.—Oklahoma and Texas.

the creation of an environment that should return the maximum in health, comfort, efficiency and pleasure to the occupants thereof. Reference to Table 1 shows that the cooling-load tonnage assigned to lighting varies from 2.3 per cent of the total in the case of a restaurant in the southwest to 47 per cent in the case of an apparel store in the northwest. As another indication of magnitude, if all of the cooling load components in Table 1 are averaged arithmetically, the results are as follows:

Table 2—Average of Cooling Load Components

	Per Cent Total Cooling Load
Fresh Air	24
Conducted Heat	24
Lighting	18
Occupants	18
Miscellaneous Heat Gains	11
Solar Radiation Through Windows	7

It is evident from a study of the ratio of lighting watts installed to watts in use, Item 11, that the demands of modern business are such that much of the available lighting is in use during periods of air cooling. In 17 of the 82 installations studied, all of the watts installed were in use during cooling periods. With one not reported, the remaining 64 averaged 72 per cent of total watts installed, in use during cooling periods, thus providing a safety factor of a low order when viewed against the background of lighting growth and practice. This is further supported by the evidence disclosed in Item 9, showing the average lighting watts per square foot to be 2.36. Ignoring variations in types of lighting systems, this indicates that generally speaking the lighting levels in the areas studied would not be expected to be any more than the minimums required, and hence enlargement of this element of the cooling load is inevitable.

However, a study of Item 8 shows that very little spare capacity is being provided to allow for this expansion. Therefore, it seems pertinent to point out to engineers and architects the desirability of carefully studying contemplated installations with a view to providing more spare capacity initially, or making provision for future expansion with a minimum of incremental cost. What this factor of safety is to be will of course depend upon the type of area being conditioned and the relationship between lighting load and total cooling load. In restaurants, for example, the lighting load appears to be less than 10 per cent of the total cooling load. Offices show a ratio of around 20 per cent, while mercantile establishments vary from 10 per cent to 50 per cent. Areas showing small percentages offer less of a problem than do those with heavier lighting loads. At the same time, areas devoted to seeing tasks and merchandising undergo the most rapid changes in lighting so that air-conditioning capacity should be thoughtfully planned, in order to safeguard the ultimate best interests of consumer and industry alike.

It is uncontrovertible that in an air-conditioned interior a given lighting result should be accomplished in the most effective manner, taking into consideration the illumination requirements, resultant influence on cooling, and comparative costs. There is first the choice of light utilization devices involving materials, cost, method of application and efficiency. Next, the commercial development of gaseous discharge lamps, including the principle of fluorescence, presents another set of alternatives to the engineer. It is increasingly possible to obtain the higher standards of illumination known to be desirable in the attainment of better seeing, with relatively smaller increments of sensible and radiant heat, but the cost of the lighting system to accomplish this result does not bear a simple relationship to the actual or theoretical reduction in the energy and equipment costs necessary to remove the heat gain. The Committee therefore presents for consideration a method whereby the cost relations between lighting and its concomitant cooling component can be analyzed. The lighting constants were taken from a standard comparison method promulgated by the Edison Electric Institute,[4] and now in general use by illuminating engineers. The air-conditioning constants are in part derived from the fundamentals of physics and from standard practices of the industry.[5] This method constitutes a formula that can be applied for the purpose of comparing the effect of any number of lighting plans upon cooling load components.

In order to develop a better understanding of its application, there is presented in Table 3 an actual study of a large drafting room now air cooled, and lighted with revamped totally indirect equipment averaging 3 w per square foot. Original air cooling capacity allowed for 5 w per square foot but the spare capacity having been diverted to other areas, the owners desired to know whether it would be more economical to install additional capacity to allow an incandescent lighting load of 5 w per square foot, or hold the lighting load at present capacity by utilizing fluorescent lamps. No conclusions should be drawn from the numerical results shown because every case must be judged in the light of conditions that exist. For the sake of simplicity only round numbers are used in the calculations.

The figures relating to lamp and fixture performance and prices are correct as of April, 1941. The basic method will, of course, be uninfluenced by changes in equipment or prices. Experience has shown, however, that several of the items should be carefully investigated in each instance.

Lamp cost and luminaire cost should be entered at the price which would be quoted through legitimate channels. Light sources, for example, may be bought on contract, or in standard package quantities over the counter, at established discount. There is no intention of advocating the publication of discounts or the purchase of equipment at other than established prices, but true values should be used.

In the example given, Item 10 on wiring cost was left blank because there is no evidence of sufficient uniformity of conditions or cost to risk the use of a figure which might cause embarrassment. It should not be left out of specific studies if additional wiring is involved. Many relighting jobs, on the other hand, may not require rewiring. However, the influence of poor power factor should not be ignored in wiring calculations.

Installation costs have not been provided for, nor has the influence of labor charges for lamp replace-

[4]Fluorescent Lighting Practice, (Published by Edison Electric Institute, December, 1939.)
[5]Heating, Ventilating, Air Conditioning Guide, 1941—Chapters 6-8-20-21-22-23-25.

Table 3—Analysis of Lighting and Air Conditioning Systems for a Large Drafting Room

A—Analysis of Lighting Systems

		System 1	System 2	System 3
1	Type of lighting	Converted Indirect	New Indirect	Semi-indirect Fluorescent
2	Number of luminaires	102	102	102
3	Wattage per unit	500 SB	500 SB	300 (with ballasts)
4	Lamps per unit	1	1	6 (48" white fluorescent)
5	Rated lamp life hours	1000	1000	2500
6	Lamp cost, each	$1.28 (List, less 25%)	$1.28	$1.20
7	Lamp renewal cost for all fixtures	$130	$130	$734
8	Luminaire cost per outlet, net	——[a]	$15	$66
9	Total luminaire cost	——	$1,530	$6,732
10	Wiring cost	Present	wiring adequate	
11	Total investment	——	$1,530	$6,732
12	Yearly taxes, insurance and depreciation (25 percent of Item 11)	——	$383	$1,683
13	Burning hours per year	2000	2000	2000
14	Annual lamp renewal cost, net	$260	$260	$588
15	Total lamp renewal cost and write-off (12+14)	$260	$643	$2,271
16	Total wattage of all lamps and auxiliaries	51 Kw	51 Kw	31 Kw
17	Kwh/year (2000 hours use)	102,000	102,000	62,000
18	Annual energy cost. Always use the actual rate that would apply to incremental load; in this case based on one and one-half cents per kwh	$1,530	$1,530	$930
19	Total yearly charges (15+18)	$1,790	$2,173	$3,201
20	Average foot-candles in service	30	36	50
21	Cost per foot-candle year	$60	$60	$64

[a] Existing fixtures already written off; substitution of silver-bowl bulbs sufficient to make them reasonably modern.

B—Analysis of Air-Conditioning Capacity Assignable to Lighting

22	Lighting load requiring additional cooling capacity, Kw	20	20	None
23	Btu per hour = Kw x 3415	68,300	68,300	
24	Tonnage of air-conditioning system = $\frac{\text{Btu per hr}}{12{,}000}$	5.7	5.7	
25	Installed cost of air-conditioning system. In this case based on $400 per ton; if exact figure not known, $300 a ton may be assumed as fair average for commercial installations; no averages applicable to industrial areas because of wide variations in design and installation requirements.	$2,280	$2,280	
26	Hours use of air-conditioning	1,000	1,000	
27	Annual power cost. Average air-conditioning system input approximately one kw per ton; power cost = air conditioning tons × hours use of compressor × energy rate.	$57	$57	
28	Hours use of water	2,500	2,500	
29	Water cost. Dependent upon water temperature, average 2.5 gpm per ton; water cost = 150 × air-conditioning tons × hours use of water × water rate. (13c per 1000 gal for this study).	$278	$278	
30	Refrigerant cost. One charge per year average, at $2.00 per ton.	$12	$12	
31	Labor and maintenance. Average $10 per ton per year.	$57	$57	
32	Taxes, insurance, interest and depreciation (15 percent of Item 25).	$342	$342	
33	Total yearly charges (Sum of Items 27, 29, 30, 31, and 32).	$746	$746	

C—Combined Lighting and Air-Conditioning Cost Summary

34	New money required			
	Lighting (Item 11)	.....	$1,530	$6,732
	Air-Conditioning (Item 25)	$2,280	$2,280	.....
		$2,280	$3,810	$6,732
35	Annual ownership cost			
	Lighting (Item 12)	.....	$383	$1,683
	Air-Conditioning (Item 32)	$342	$342	.....
		$342	$725	$1,683
36	Annual operating costs			
	(a) Lighting			
	Lamp cost (Item 14)	$ 260	$ 260	$ 588
	Current cost (Item 18)	1,530	1,530	930
		$1,790	$1,790	$1,518
	(b) Air-Conditioning			
	Power (Item 27)	$ 57	$ 57	.....
	Water (Item 29)	278	278	.....
	Refrigerant (Item 30)	12	12	.....
	Maintenance (Item 31)	57	57	.....
		$404	$404	
37	Total yearly charges			
	Lighting (Item 35)	.....	$ 383	$1,683
	(Item 36a)	$1,790	1,790	1,518
		$1,790	$2,173	$3,201
	Air Conditioning (Item 35)	$ 342	$ 342	.....
	(Item 36b)	404	404	.....
		$746	$746	.....
	Total	$2,536	$2,919	$3,201
	Total per foot-candle year	$ 84	$ 81	$ 64

ments been included. These may be sizable items, particularly in an installation involving a multiplicity of light sources, and should be considered if figures are available.

Items 12 and 32 are ones over which some difference of opinion may arise. It is urged that when preparing an analysis for a specific job, the fixed charge percentage be agreed upon with the customer. Some may prefer to spend money and forget it. Others may wish to use a different rate of depreciation for wiring than for fixtures. Insurance and taxes will vary. However, the fixed percentages used in the example represent good accounting practice and may be used in the absence of specific data.

Power costs will, of course, vary with the locality, the magnitude of use, and the class of service. In smaller installations, a variation in consumption between two methods of lighting may result in per kwh charge differences, because of the effect of sliding scale or block rates, or as result of demand component in two-part rates.

The hours use of water may not coincide with the hours use of a cooling compressor. Under certain conditions, water alone may be used during certain periods to effect a reduction in air temperature, thus adding to the total hours during which water is consumed. In this particular study, water was used for 2500 hours a year, compared with 1000 hours for the compressor.

A growing practice is the use of cooling towers or evaporative condensers which reduce the amount of water required to around 4 per cent of that consumed by the *once-through* use of city water. However, this results in an increase in the total investment and use of auxiliary power, all of which would be resolved in arriving at the items of over-all water cost.

Item 25, Installed Cost of Air-Conditioning Equipment, is subject to considerable variation. In many instances, *additions* to an existing system would involve less in the way of control devices and engineering expenses, thus reducing the amount per ton to perhaps as low as $250.00. It is often the case that the exact amount of additional capacity to handle the lighting load cannot always be purchased. In the study in question, a compressor of

5.7 tons would not be purchased: a 6-ton unit would be standard. On the other hand, when installing additional incremental capacity there may be other cooling requirements to consider which would allow the exact amount needed for lighting to be allocated to it for accounting purposes.

KUGEL HEADS SMOKE PREVENTION ASSOCIATION

Delegates attending the 35th Annual Convention of the *Smoke Prevention Association of America,* held recently in Atlanta, elected the following officers to serve for the ensuing year: H. K. Kugel, Washington, D. C., *President;* W. E. E. Koepler, Bluefield, W. Va., *First Vice-President;* S. F. Peters, Chicago, Ill., *Second Vice-President;* Frank A. Chambers, Chicago, Ill., *Secretary-Treasurer;* and Andrew Jones, Atlanta, Ga., *Sergeant-at-Arms.*

GOVERNMENT BUYING RULES

A pamphlet designed to supply information concerning procedures employed by various Government agencies in making routine purchases of merchandise for civilian needs has been made available by the Department of Commerce for the information of businessmen who desire to sell to Federal units.

It is not likely that the purchasing methods by the various agencies of the Government as outlined in the booklet will be radically changed by the impact of defense needs, it was announced.

There is no great mystery involved in doing business with the Government, according to the Department of Commerce. Any businessman desiring to do such business may write to the Procurement Division, Treasury Department, Washington, D. C., and request to be listed to receive invitations to bid on items he can furnish.

Army and Navy purchasing is explained by two booklets published by the respective departments. The *Army Purchase Information Bulletin* (1940) may be had by writing the War Department, Washington, D. C. *Selling to the Navy* may be had upon application to the Department at Washington or any of its field offices in the United States.

President's Message to Members

Business analysts have pointed out that progressive industrial expansion continues to be an outstanding characteristic of 1941 business, and it has been forecast that the yearly average of business for 1941 will reach the highest point of production for the past one hundred years. This serves to emphasize some of the pressing engineering problems connected with comfort, efficiency and the health of the public, which need solution. Industrial operations on a 24-hour working basis put a severe strain on physical stamina, and the expansion of industry for defense has created a housing problem, with its accompanying public health problems. The enforcement of priorities for materials makes the subject of substitutes highly important.

Uncle Sam is a big customer for the period of the emergency, but what about the years to come? At no previous time have there been so many problems confronting the heating, ventilating and air conditioning profession, and they are of pressing importance right now.

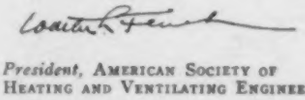
The Society's 48th Annual Meeting is to be held in Philadelphia, January 26 to 29, 1942, and it will provide a suitable forum for the discussion of those topics which involve public health, efficiency and comfort. Recognition of the Society's leadership in Research and in its allied fields of activity has been given by government departments, which require immediate answers to important defense problems. For these investigations, much of the research capacity of our laboratories has been assigned.

Unlike those meetings which are held for entertainment and the advancement of social contacts, the Society's Philadelphia Meeting will be strictly business, devoted to discussions of major research and engineering problems in the fields of heating, ventilating and air conditioning. The Committee on Arrangements of the Philadelphia Chapter, which is acting as host, plans for the biggest meeting ever held by the Society.

Because heating, ventilating and air conditioning are essential industries, they enter into every industrial and public building program. For the army camp or home to maintain comfort, efficiency and the health of the occupants, heating, ventilating and air conditioning must play an important part.

For this reason, the exhibits of manufacturers at the 7th International Heating and Ventilating Exposition, to be held at the Commercial Museum in Philadelphia at the same time as the Society's meeting, take on increasing importance. Much of the current production of goods has been allocated to the defense effort. Normal buying and selling methods have been interrupted, but future needs must be recognized and policies established for the ultimate resumption of established services. The manufacturers who participate in the Philadelphia Exposition will find an eager and interested audience for their exhibits of new and better equipment for the plants and homes of the nation.

Therefore, I feel that we should back the Philadelphia Meeting and the Philadelphia Exposition with re-doubled effort and enthusiasm and without a thought of curtailment.



President, American Society of Heating and Ventilating Engineers

September 1, 1941.

1941-42 Chapter Officers

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Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 10 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by September 15, 1941, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BOOTH, CLIFFORD A., Sales Engr., Fiberglas Canada Ltd., Montreal, Que., Canada.	W. U. Hughes G. L. Wiggs	F. G. Phipps N. J. Forrester
GUNTHORPE, CHARLES E., Mgr., Heating Equip. Dept., Campbell Coal Co., Atlanta, Ga.	F. L. Laseter C. B. Cole	A. H. Koch L. F. Lawrence, Jr.
HARE, WILFRED A., Executive Partner, Sawyer-Hare Furnace Co., Detroit, Mich. (*Reinstatement*)	J. F. McIntire R. F. Connell	J. S. Kilner E. H. Clark
HECKEL, EDMUND P., JR., Engr. Buffalo Forge Co., Buffalo, N. Y.	A. A. Criqui W. R. Heath	C. A. Booth R. D. Madison
NORTON, L. IVAN, Sanitary & Htg. Engr., Eveready Plbg. & Htg. Co.	A. L. Walters C. W. Helstrom	R. H. Ahrens T. R. Johnson
RAISLER, ROBERT K., Treasurer, Raisler Corp., New York, N. Y. (*Advancement*)	L. K. Berman H. A. Adams	R. H. Carpenter W. H. Driscoll
SHEFFIELD, RAYMOND A., Owner, Air Conditioning Engrg. Co., Cambridge, Mass. (*Reinstatement*)	H. M. Tarr C. P. Yaglou	C. M. F. Peterson D. J. Edwards
SMITH, GEORGE E., Supervising Engr., Pinner, Middlesex, England.	G. F. Hafferfield (*Non-member*) S. U. Stevens (*AISE*)	A. L. Moon (*AISE*) C. Allen (*Non-member*)
SNYMAN, G. C., Regional Mgr., Carrier Corp., Syracuse, N. Y.	W. H. Carrier L. L. Lewis	Margaret Ingels C. M. Ashley
YERKES, WILLIAM L., Engr., Carrier Corp., Philadelphia, Pa. (*Reinstatement*)	W. A. Bornemann W. A. Grant	E. H. Dafter A. P. Shanklin

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ATKINS, GEORGE E., Consulting Engr., San Francisco, Calif.

BLAIR, ERNEST L., Industrial Engr., Stone & Webster Engrg. Corp., Boston, Mass.

BOLAND, L. C. JR., Consulting Engr., Atlanta, Ga.

DONOHOE, CHAS. F., Engr., The Detroit Edison Co., Detroit, Mich.

HARBIN, FRANK JR., Engr., Home Furnace Co., Holland, Mich.

HUMES, W. EARL, Contractor, Reno, Nev.

HUNTER, THOS. B., Consulting Engr., Hunter & Hudson, San Francisco, Calif.

MILLER, JOHN W., Engr., Motor Wheel Corp., Lansing, Mich.

OWEN, CHARLES E., Mfr's. Agent & Distributor, Carbondale, Ill.

POUND, HOWARD W., Mgr., Electro-Matic Div., American Air Filter Co., Louisville, Ky.

REESE, HENRY L., Consulting Engr., Reese & Co., Reading, Pa. (*Reinstatement*)

SANDER, ANDY J., Engr., Chicago Master Steamfitters Assn., Chicago, Ill.

STEVENS, HOWARD R., Engr., H. E. Savier & Son, Inc., Reno, Nev.

TILFORD, LEO A., Owner & Mgr., Leo A. Tilford Co., Jackson, Mich.

WOLFE, JOHN S., Acting Architect, Mech. Engrg., Milwaukee School Board, Milwaukee, Wis.

ASSOCIATES

BARRETT, CAMPBELL M., Mgr., General Equipment Co., Ltd., Vancouver, Can.

CARROLL, DANIEL E., Pres., Carroll Sheet Metal Works, Inc., Woodside, L. I., N. Y.

DOVENER, ROBERT F., Engr., Atchison & Keller, Washington, D. C.

FINDLEY, HOWARD N., Mgr., Avery Engrg. Co., Cleveland, O.

MEAD, GEORGE E., Sales Engr., George E. Mead Co., Seattle, Wash.

MEILLER, DANIEL V., Gas Testing Engr., Public Service Co. of North Illinois, Oak Park, Ill.

PACKTOR, BERNARD M., Chief Engr. & Pur. Agt., Air Distributors, Inc., New Haven, Conn.

STEVENS, JUDSON E., in charge Htg. Engrg. Dept., National Coal Co., Reno, Nev.

WALKER, WYTHE F., Engr., Hayes Furnace & Mfg. Co., Los Angeles, Calif.

JUNIOR

LEONARD, R. R., Sales Engr., A. M. Byers Co., Philadelphia, Pa.

STUDENT

WEARANGA, ROBERT R., Student, Michigan State College, East Lansing, Mich.